CRYOGENIC CARS

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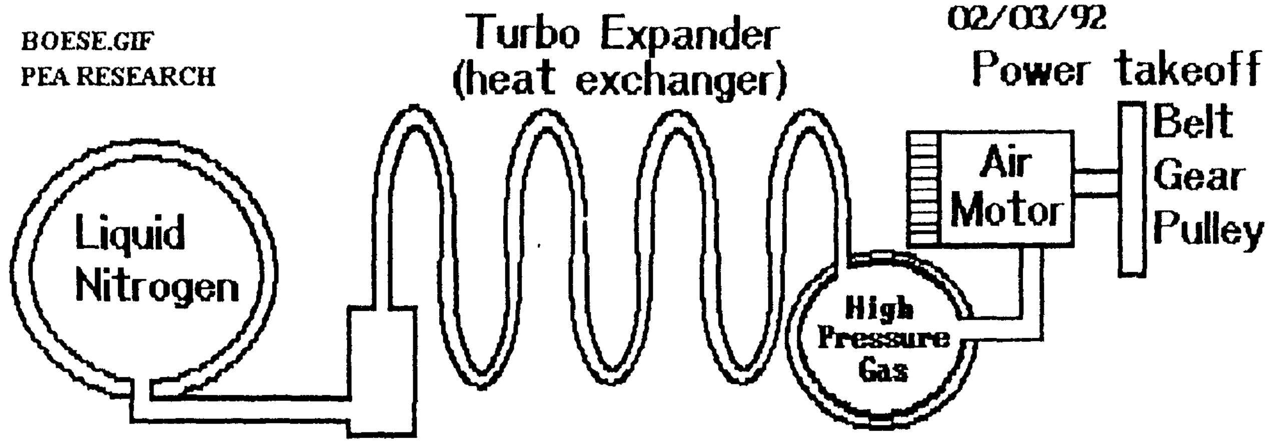
BOESE NITROGEN MOTOR (BOESE.GIF)



The Liquid Nitrogen Motor takes advantage of the expansion of liquid nitrogen to a gaseous phase. This entails a volume increase of 980 times. One gallon of Liquid Nitrogen equals 231 cu. inches which converts to 226,380 cubic inches as nitrogen gas.

Liquid Nitrogen is stored in a DeWar flask (like a coffee thermos). When the liquid nitrogen moves through a series of warmer conduits the ambient heat causes the liquid to expand by 980 times to a now high pressure, non-polluting gas that then provides a motor force. Nitrogen gas comprises 78% of the Earth's atmosphere, is inert and totally non-polluting. Liquid Nitrogen will last for up to 1 month in it's liquid state without extra insulation...total cost 2-5 cents per mile

SECRETS OF THE AIR CARBURETOR

PATENT No.
N/A

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File Name : BOESE.ASC | Online Date : 09/22/94
Contributed by : Jerry Decker | Dir Category : ENERGY

The following file refers to the late cryogenics genius, Harold L. Boese (pronounced BAY-ZEE). He was a recognized expert in the field of super cold and has been called the Father of Cryogenics. Mr. Boese died about 4 years ago (1988) after several strokes, according to his daughter. Due to the severity of the strokes, Mr. Boese was restricted to a wheelchair in his later years. You might also take the file BOESE.GIF which shows a diagram of how the system works.

Mr. Boese developed a most unique engine based on liquid nitrogen (N2) which is inert, non-flammable and comprises about 78% of our atmosphere. A Pinto station wagon was converted to run on his engine and proven as a viable means of transport at extremely low cost and absolutely no pollution.

The engine uses a "turbo-expander" which functions as a heat exchanger. It allows the liquid nitrogen a means of rapid expansion from heating due to ambient temperatures of >30 degrees F. The "turbo-expander" is basically a long hollow copper pipe that is either spiralled or folded in such a way as to give the greatest surface exposure.

As the liquid nitrogen passes through this expansion tubing, the ambient heat will cause the liquid to change to a gas through a phase change. Liquid nitrogen expands by 980 times its volume when changing from a liquid to a gas.

Expansion means pressure which translates to power.
1 gallon = 231 cubic inches

Expansion of one gallon of Liquid Nitrogen into Nitrogen Gas
980 X 231 = 226,380 cubic inches

Once the liquid has expanded by 980 times its volume, it is fed into an off- the-shelf AIR MOTOR. An air motor is kind of like a paddle wheel on an old style riverboat. As compressed air is blown against the paddles, a shaft is spun which drives whatever is attached, in this case, the transmission of a vehicle. Boese said that only a slight modification to the air motor was necessary to make the it run perfectly with the expanded and very cool nitrogen gas.

Mr. Boese's daughter told me in a phone conversation that her dad had designed and built a liquid nitrogen motor for use as a backup engine on the NASA Lunar Rover. An electric engine WAS INTENDED as the primary with the liquid nitrogen engine as backup. I was told the N2 engine was ACTUALLY USED in the pictures we see of the Lunar 'buggy' bouncing across the surface because it was smaller, less heavy because it did not have all those batteries (despite the 1/6th earth gravity) and much more efficient.

Boese's daughter said that he was ecstatic with this success and that NASA expressed an interest in promoting this as an excellent engine for vehicles. That lasted until the mission was over and then they changed their tune and refused to support any claims that a liquid nitrogen motor was actually used.

For the remainder of Boese's life, he tried to promote and market the use of his invention as a non-polluting motive force, with no appreciable success. A company was formed, Cryogenics Unlimited Corporation, for that purpose and offers were being considered from Japan and Italy. As of this writing (originally 1992), no commitment had been arranged from anyone.

The current status of the working vehicle is that one of the financial backers has the Pinto test car in his garage. It has not worked for several years. He has been tinkering with it to find out how to make it work but does not have the late Mr. Boese's understanding of the process.

Liquid nitrogen was available for about 2 to 5 cents per mile back in the 70's when the tests were being carried out. Boese claimed he had also discovered a means of generating his own liquid nitrogen AS THE CAR WAS RUNNING, so that it could partially or fully replenish its own fuel.

The beauty of his engine is that it could be used for many other purposes than just running a car. It could drive a generator to produce power for your home among other uses. There is much to yet be discovered (or RE-discovered) with regard to this principle and it needs to be shared with the greatest number of interested people in hopes they can or WILL do experiments that will be shared with the network.

Good luck! ....Jerry

For more information on "FUTURE CARS" write to:

PEA RESEARCH
5050 LAGUNA BLVD. #112-474
ELK GROVE, CA 95758

YOU WON'T HAVE TO WORRY ABOUT TUNE-UPS WITH A CRYOGENIC CAR
From Left, Clark & Harold Boese, Bob Leski, David & Fred Ewald

Inventor has high hopes for his nitrogen-powered car in Galveston

By LAURA LAMBETH
News Staff Writer

He says nobody believes him. And when you look under the hood of the automobile you can understand why.

There's almost nothing there. No engine, no radiator, no fan belts, no transmission or motor—just a battery and a strange-looking device called a turbo expander that looks like air-conditioner coils. You won't find a gas tank, either.

And the car's lubricant isn't Texas tea—it's plain old salad oil.

It's the super-cool car of the future that offers more than a solution to the energy shortage.

The car is powered solely by nitrogen, the "ultimate" solution to the energy shortage, he said.

The updated design doesn't require electrical power, and about the only part it needs is a small cryogenic motor attached underneath the rear-end of the vehicle. The motor costs about $20 to rebuild in less than 45 minutes, and it's guaranteed to last at least [illegible] years without spark plugs, wiring or other accessories.

Liquid nitrogen, which is kept in the car's special tanks at a cool 320 degrees below zero, enables the motor to be thermostatically operated at minus 220. The super-cooled gas, after it passes through the motor casing, flows through a "turbo expander," to a master coil with additional turbos, where it increases its volume 720 times into a volume tank.

The finished gas is then passed harmlessly into the atmosphere, which is [illegible] quarters comprised of nitrogen, thus making the car totally pollution free.

And liquid nitrogen is non-combustible, so there's no danger of fires or explosions from the automobile's tanks, making the car safer than anything being driven on today's highways.

An added luxury is air-conditioning without a compressor, and a refrigerator that can be installed for a bar or whatever, easily cooled by the liquid nitrogen flowing through the tanks and tubing. Boese said liquid nitrogen is available at every major truck stop in the country, and it would only take five and a half hours to set a tank up at a gasoline station. It costs about one cent a gallon to manufacture liquid nitrogen and probably would sell at 14 cents a gallon before taxes.

The converted automobile gets about 20 miles to the gallon, but after Boese completes his designs, he expects a new vehicle will drive on at [illegible] [illegible] gallon. Right now, he has a 40 gallon tank hooked up to the back of a Pinto, but a redesigned car simply would carry tanks underneath the rear end.

[illegible] of designs created by Harold L. Boese, president of Cryogenics Unlimited Corporation.

Boese, a cryogenics expert in the science of super cold phenomena, worked with a partner in the early part of the 70's to build a nitrogen-cooled electrically-powered foreign compact.

But the car never drove out of Detroit Boese said because he was told it would take at least 10 years to bring it out on the market.

Boese went back to his work as an electrician, but kept his ideas and theories in his head.

Now he's back to work on his designs after receiving a request from government officials to help find a

Beoese said be doesn't plan to convert cars to cryogenic power because, once converted, the car would have a lot of useless space under the hood. "I'd have to throw away a lot of engines," he said.

His hopes are to design what he calls a "hybrid" vehicle, unconventional by today's standards, that would sell for about $4,000 on the market.

His dream is to build a manufacturing plant in the Galveston area if he can get enough financial backing through interested parties or the government. If all goes well with testing and designs, a car could be out on the roads in three years, he said.

"They'll have an electric car available by 1985. It goes 50 miles at 40 miles per hour, or vice-versa," he said. "Then it sits for 12 hours to recharge.

"So who wants that?" Boese said.

Boese [illegible] [illegible] have been working on the mechanical designs of the car, and Fred [illegible]. Ewald, Boese's partner, has provided the financial backing that has made the project possible.

The automobile sits in the yards at A-1 Rentals, friends of the company who provide security.

Boese and Ewald plan to show the car at Expo '80 in St. Louis this May.

Boese holds 78 patents in cryogenic designs, participating in the design of the car that was driven on the moon, as well as the freezing of the first human beings.

He virtually taught himself the concepts of cryogenics when he read an article on the subject 20 years ago while working in the Frozen Right Products Division of Burris Mills Inc. in Dallas.

He said he believes cryogenics, currently an undeveloped field, may be an up and coming area of study as an alternative energy source in the future, since atmospheric gases are easily accessible.

The cryogenic car is one project [illegible] [illegible] to talk about, but he has other ideas he's keeping quiet right now, which he'll be showing in the near future, that he promises will be "real interesting."

Take a ride in his super-cool cryogenic car and maybe you'll believe him.

The Boese auto is simply his spl little turbine desighn attached to the rear end driveshaft of a Ford Pinto. A number of Companys have been interested in developing it and putting it on the market but the inventor has held off on his decisions after his Ist model was 'accidentally' blown up by individuals who had purchased it. (3½ million) Some of the major desighn secrets were withheld from the more than 70 patents covering the desighns••• 1980 -

U.S. Patent 3,842,333, 10/74

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Take a ride in his super-cool cryogenic car and maybe you'll believe him.

Coming soon—free electric power for all America's homes

AMERICA COULD be standing on the brink of the greatest economic boom in all history thanks to the invention of a Nevada engineer.

The invention could solve the world's energy problems, create tens of thousands of new jobs, save giant American industries threatened by foreign competition, and, incredibly, provide American homes with all the electricity they need — absolutely free!

• CHARLES DARLING reveals company's plans

In an exclusive interview Charles Darling, president of Quantum Energy, Inc., revealed to the EXAMINER his company's plans to have the invention, a home energy unit capable of providing free energy to homeowners, in production within the next 58 weeks.

"The prototypes have been thoroughly tested and approved by top engineers," Darling said. "We have just completed negotiations with Merlin Technology, Inc., a subsidiary of the giant Celanese Corporation, to produce a production prototype.

"Within 58 weeks they'll give us the model we need to go into full production in the plant we're getting ready to build in Houston, Texas," an excited Darling said.

"The technology is really quite simple. The home energy unit gets its energy from low grade ground heat which is available two to three feet below ground level anywhere in the world.

"This ground heat drives the generator which is sealed inside a self-contained unit. The energy produced will provide all the power needed for the average home for heating, air conditioning, cooking, hot water, and lighting.

"In short, the unit will eliminate the need for any other source of energy. Moreover since it's contained within the home, there will be no problems of power failures or breakdown."

Darling told the EXAMINER that the cost of the home units would be less than $6,000, will last from 25 to 30 years and will carry a 20 year guarantee. In the event of a malfunction, dealers will replace the unit at no cost to the owner, within 24 hours of the breakdown.

The units will be pollution-free, almost noiseless, and sealed inside a cabinet roughly 40 inches long, two feet wide, and 30 inches high.

Robert Torczyner, general manager of Merlin Technology Inc., confirmed that his company is now working under contract with Quantum Energy to produce the prototype. He told the EXAMINER: "It's a very exciting project indeed."

— DAVID GUARDINO

Liquid Nitrogen as Alternate Energy

Demonstration car carries cryogenic liquid fuel in vacuum-insulated tank. Released gas expanded in front-end evaporator powers simple air motor. Such vehicles are planned for underground mines and other areas of high explosion risk

Britain's National Coal Board is working on small vehicles powered by liquid nitrogen for transport in underground mines, where the non-flammable non-toxic fuel has an important safety advantage. This follows successful application for the inert gas to operate standard pneumatic tools.

The system uses normal ambient air temperature to heat and vaporize the nitrogen, which boils at -196 deg C. The gas then expands like steam and can drive an air motor. In this basic form no additional heat source is needed.

The experimental vehicle pictured has a vacuum-insulated storage vessel in which the cryogenic liquid is held at the boiling point. Pressurized vapor is released from the tank and piped through the front-end heat-exchanger where it is warmed and expanded. Gas at 100 psi is then fed to a conventional 2.75-hp four-cylinder non-expansive air motor driving the wheels. The car has a maximum speed of 10 mph, and has been used to demonstrate the feasibility of this energy source for light self-propelled units.

Low bulk gives liquid nitrogen a major advantage over compressed air for automotive applications, since it produces six times the volume of expanded gas for a container of the same size. It is easily stored, and a pressurized vessel is unnecessary, which means much lower weight.

One volume of liquid expands to 640 volumes of gas at atmospheric pressure, while for an equal expansion from a compressed gas a storage pressure of about 9400 psi would be needed. Compared with other non-fossil portable energy sources the liquid has a much higher storage capacity in terms of hp hr. lb than even silver-zinc batteries.

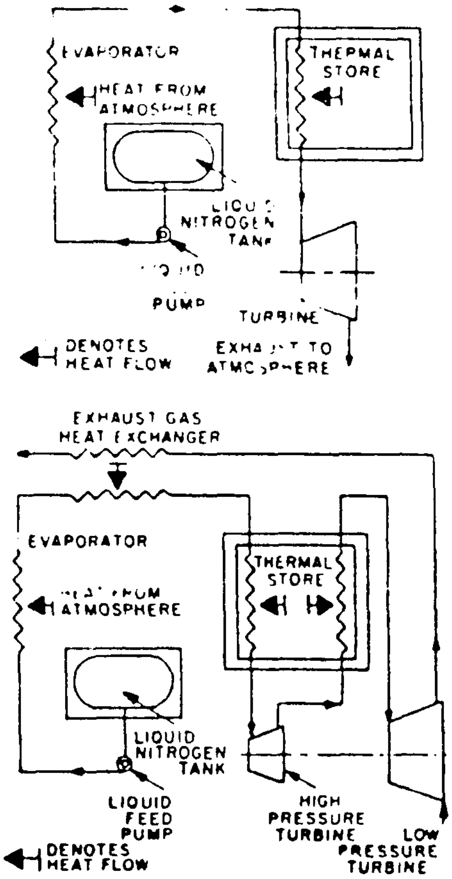
The Coal Board is cooperating with Salford University, near Manchester, which has done considerable theoretical and experimental work on improving the performance of liquid nitrogen engines for road transport. A principle area of study is the use of a thermal store to raise the temperature and thus the expanded volume of the gas fed to the engine. This could dramatically reduce consumption of the fluid for a given work output, allowing greater power or increased range, or a smaller on-board liquid supply.

Possible thermal stores, heated overnight by off-peak electricity, could be solid refractory materials like graphite or alumina rods or bricks through which the gas is passed. Others are eutectic salts using the latent heat of fusion, and units relying on chemical reaction such as the catalytic oxidation of a fuel.

Salford's studies are concerned with turbines rather than reciprocating engines, and these envisage applying the warm exhaust gas to the primary evaporator following the liquid container to raise the heating temperature above ambient. A more sophisticated system would have two-stage expansion with high- and low-pressure turbines, the inlets to both passing through a common heat sink.

Further possibilities for vehicles include a regenerative braking system where the heat developed by friction would be recovered and held in the store. Liquid nitrogen for propulsion is also seen as particularly suitable for refrigerated trucks, when the cargo compartment could be readily cooled by a special evaporator in the heat-exchanger circuit. It would then contribute to the power source instead of drawing on it.

Nitrogen gas is produced commercially in liquid form by distillation from the atmosphere. The liquid is now readily available in bulk relatively cheaply. It is used widely in the chemical industry, steel making, and freezing equipment.



Thermal store in gas expansion circuit could increase efficiency of liquid nitrogen engine. Lower drawing shows two-stage turbine, with exhaust providing additional primary heat step-up. Concepts are patented by Salford University.

INVENTION

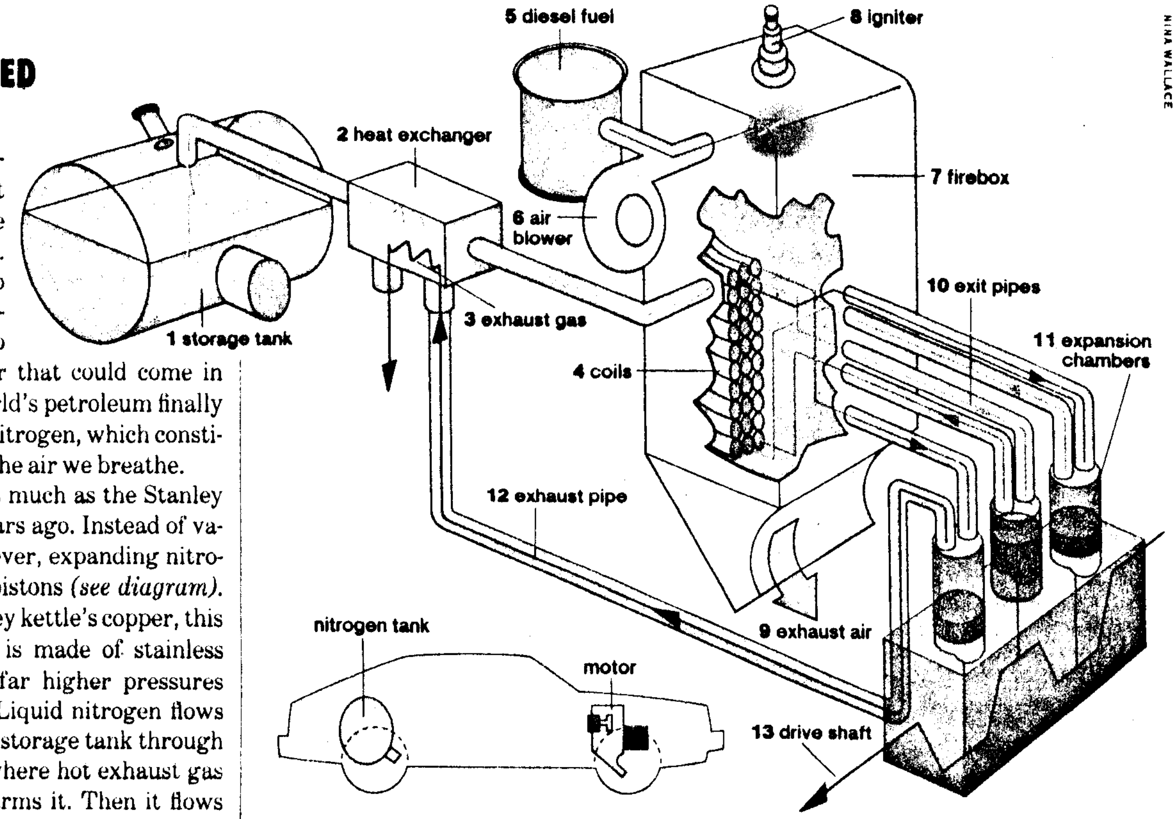
AIR-POWERED AUTO

Tinkerers are forever trying to invent something to replace the gasoline engine. The latest are two California automotive engineers who have designed a car that could come in handy when the world's petroleum finally runs low. Its fuel is nitrogen, which constitutes 80 per cent of the air we breathe.

The engine works much as the Stanley Steamer's did 87 years ago. Instead of vaporized water, however, expanding nitrogen gas drives the pistons *(see diagram)*. In place of the Stanley kettle's copper, this high-tech variation is made of stainless steel to withstand far higher pressures and temperatures. Liquid nitrogen flows from a thermos-like storage tank through a heat exchanger, where hot exhaust gas from the engine warms it. Then it flows into a firebox, which burns a small amount of diesel fuel. The heated nitrogen expands as it changes to gas, creating pressures between 3,000 and 3,500 pounds per square inch and temperatures above 1,800 degrees. The nitrogen passes back and forth from three coils to three cylinders, which turn a shaft that transmits power to the wheels.

The designers, James Dooley and R. Philip Hammond of R & D Associates in Marina del Rey, financed their original research with Department of Energy money. With plans in hand, the team is seeking backers for a $5 million prototype.

The new hotrod would create little pollution, make hardly a sound, and cost about as much as a conventional automobile. It should get about four miles to the gallon of liquid nitrogen—costing an estimated ten cents a gallon—and could travel 300 miles between fillings of its 85-gallon tank. But an entire new infrastructure of service stations would have to be built to supply the nitrogen fuel—cooled to a chilly minus 390 degrees—before the air-powered car could hit the road.



NINA WALLACE

Nitrogen flows from a vacuum-insulated storage tank (1) through the heat exchanger (2) to be warmed by still-hot exhaust gas (3). Next, the gaseous nitrogen, traveling through the first of three sets of coils (4), is heated further by the burning of diesel fuel (5), forced by an air blower (6) into the firebox (7), and sparked by an igniter (8). Exhaust air (9) from the firebox exits from the bottom. The hot, expanding nitrogen moves through exit pipes (10) between coils and expansion chambers (11). It then flows through the exhaust pipe (12) and back to the heat exchanger. Pistons transmit power to the drive shaft (13).

OLD DOG, NEW TRICK

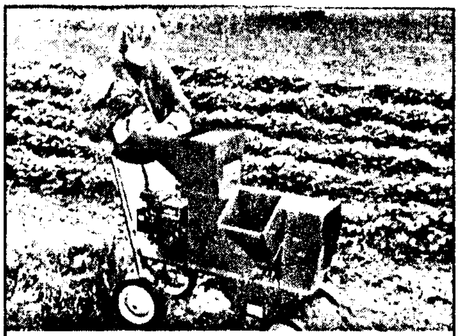
For the owner of a dog that just won't stop yapping, its bark *is* worse than its bite. Neighbors complain, fines for disturbing the peace are levied. Now there's a solution to the problem: a $200 electronic collar, available from Arizona-based Tri-Tronics, that produces an audible buzz and a mild, harmless electric shock whenever it detects a bark.

CYNTHIA MATTHEWS

An electronic collar silences chronic barkers.

According to its creator, Daniel Tortora, a psychologist and animal trainer, the collar is based on classical Pavlovian conditioning. To make behavioral training effective, says Tortora, "you have to discipline the dog every single time unwanted barking occurs." Inconsistent punishments merely confuse an animal, and badly trained dogs often resort to "sneak barking" when left alone.

In studies over the past three years, Tortora has amassed more than 8,500 dog-days of training with the collar. His results are nothing to sniff at: a habitual barker learns to be silent after wearing it an average of just 30 minutes. Tortora has also found that it takes only about three days of reinforcement training to make the new trick stick. To ensure that a dog doesn't start barking again, it can continue to wear either the real collar or a lookalike dummy. *Tom Dworetzky*



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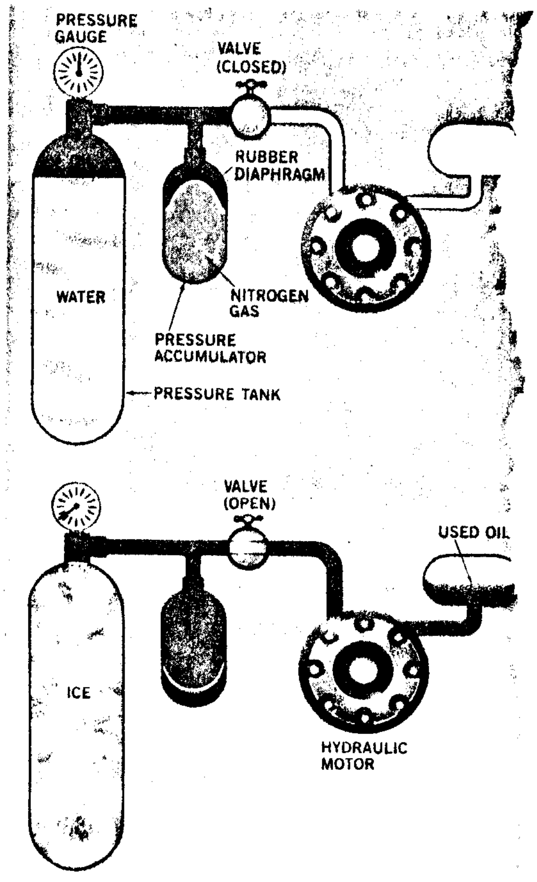
Ice power

Icy Rider uses a 27-liter oxygen cylinder as a pressure tank. With a stronger tank, the inventor hopes its driving range will increase to several miles.

As freezing water expands, its force can fracture huge rocks and burst steel pipes. To tap this ice power Bo Nordell, a researcher at the Lulea University of Technology in northern Sweden, built a small cartlike vehicle he calls Icy Rider.

Nordell uses a water-filled pressure tank to transfer energy to an accumulator, a small cylinder filled with nitrogen gas. As the water freezes, the accumulator stores the pressurized fluid that powers a hydraulic motor (see drawing). "Leave the cart outside overnight in our Swedish winter, and in the morning I can drive it at fifty miles per hour for a few hundred yards," he told me.

The on-board pressure tank could be left at a base station, charging the storage accumulator between runs. This would halve the present vehicle weight to 130 pounds. Nordell's ice power not only taps a new form of renewable energy, but is silent and nonpolluting.—*David Scott*



DRAWINGS BY BOB LANGE

Pressure tank is filled with nine-tenths water and one-tenth oil (top). Freezing water expands (right), feeding oil into the accumulator, compressing the nitrogen gas. Valve releases pressurized fluid to drive the hydraulic motor.

UNIVERSITY of NORTH TEXAS **NEWS** SERVICE

08/12/97
For release:

CooLN2Car: A car that runs on air

DENTON (UNT), Texas -- Air. Its life-sustaining property and fresh, abundant supply were taken for granted until the birth of the industrial age and its resulting offspring: the modern, smog-producing automobile. Today children learn early about car exhaust, air pollution and ozone alerts. Tomorrow they'll learn about how a prototype car, which ran on air instead of fossil fuel, reshaped their future.

On July 30, an interdisciplinary team of scientists from the University of North Texas presented CooLN2Car, a nitrogen-powered, zero-emission vehicle, at the Cryogenic Materials Conference in Portland, Ore. On Monday, Aug. 18, at 2 p.m. (CDT), researchers will present a 30-minute public information briefing followed by a vehicle demonstration at UNT.

The radically new energy concept involves the use of a cryogenic heat engine to power a vehicle using either liquefied air or liquid nitrogen. CooLN2Car gets its power from the pressure released when "cooled" liquid nitrogen is heated by outside air and becomes a gas.

Liquefied air is produced by taking air from the atmosphere, cooling and compressing it. Liquid nitrogen is similarly produced using nitrogen gas. Because air consists of 78 percent nitrogen gas, liquefied air and liquid nitrogen have similar properties.

In order to run a cryogenic heat engine using liquefied air, the liquefied air is heated and expanded in much the same way as water when used to run a steam engine. However, an important difference is that pollution-emitting combustion is unnecessary for the heating process. Instead, heat already existing in the atmosphere is used. As a result, the exhaust consists solely of pure, clean, breathable air.

The concept of applying existing liquid-nitrogen technology to power an automobile was published by UNT's Dr. Carlos Ordonez in a 1996 American Journal of Physics article, "Cryogenic Heat Engine."

UNT's research was funded by a grant from the Texas Advanced Technology Program. Co-investigators of the project are: Dr. Ordonez of the physics department, Dr. Mitty Plummer of the department of engineering technology and Dr. Rick Reidy of the department of materials science. For more information, see the web site at http://www.mtsc.unt.edu/CooLN2Car.html.

UNT News Service Phone Number: 940/565-2108
Contact: Chris Smith 940/565-4644
Email Address: csmith@abn.unt.edu

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Cryogenic Heat Engine as a Zero Emission Vehicle

Picture this: The more cars are driven, the cleaner the air becomes. Sound crazy? It's possible by using a C-H ("cryogenic heat") engine to power cars. A C-H engine is a radically new energy concept being investigated by Drs. Carlos A. Ordonez, Mitty C. Plummer, and Richard F. Reidy at the University of North Texas with support from the Texas Advanced Technology Program. The most intriguing aspect of the concept is that the fuel used by a C-H engine to power a vehicle can be either liquefied air or liquid nitrogen. Liquefied air is produced by taking air from the atmosphere and cooling and compressing it. Liquid nitrogen is similarly produced using nitrogen gas. Because air consists of 78% nitrogen gas, liquefied air and liquid nitrogen have similar properties. In order to run a C-H engine using liquefied air, liquefied air is heated and expanded in much the same way that water is heated and expanded when used to run a steam engine. However, an important difference is that pollution-emitting combustion is not necessary for the heating process. Instead, heat already existing in the atmosphere is used. As a result, the exhaust consists solely of pure, clean, breathable air. In order to prove that the concept is possible, an automobile has been converted for operation using a C-H engine. In the next phase of the planned research effort, a C-H engine electric power system will be developed and used to power an electric vehicle. The electric vehicle will be designed to travel 130 miles on a single tank of fuel. Assuming that solar energy is used to produce the fuel, the cost of fuel for operating the electric vehicle can potentially reach less than 10 cents per mile. As an environmental benefit, liquefied air and liquid nitrogen production systems remove undesirable airborne pollutants from the air. As a result, the more that cars powered by C-H engines are driven, the cleaner the air becomes.

More information can be found in our papers:

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Website maintained by R.F. Reidy and last updated December 17, 2001.

Cryogenic Heat Engine Research Technical Articles
(see also: http://www.mtsc.unt.edu/CooLN2Car.html)

"Cryogenic Heat Engines for Powering Zero Emission Vehicles," C. A. Ordonez, M. C. Plummer, and R. F. Reidy, Proceedings of the 2001 ASME International Mechanical Engineering Congress and Exposition, Paper No. IMECE2001/PID-25620.

"Liquid Nitrogen Fueled, Closed Brayton Cycle Cryogenic Heat Engine," C. A. Ordonez, Energy Conversion and Management **41** (2000) 331-341.

"A Review of Liquid Nitrogen Propelled Vehicle Programs in the United States of America," M. C. Plummer, C. A. Ordonez, and R. F. Reidy, Proceedings of the International Scientific Conference on Automobile Transport and Road Enterprises on the Threshold of the Third Millennium, Paper No. 629.1 (2000) 47-52.

"Liquid Nitrogen as a Non-Polluting Vehicle Fuel," M. C. Plummer, C. A. Ordonez, and R. F. Reidy, Proceedings of the 34th Intersociety Energy Conversion Engineering Conference, SAE Paper No. 1999-01-2517 (1999).

"Cryogenic Heat Engine Experiment," M. C. Plummer, C. P. Koehler, D. R. Flanders, R. F. Reidy, and C. A. Ordonez, Advances in Cryogenic Engineering **43** (1998) 1245-1252.

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"Cold Thermal Storage and Cryogenic Heat Engines for Energy Storage Applications," C. A. Ordonez and M. C. Plummer, Energy Sources **19** (1997) 389-396.

"Cryogenic Heat Engine," C. A. Ordonez, American Journal of Physics **64** (1996) 479-481.

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University of North Texas

Frequently Asked Questions about cryogenic heat engines and our nitrogen-powered car

What is a cryogenic heat engine?
This engine uses very cold substances to produce useful energy.

How does this nitrogen-powered car work?
Heat from the atmosphere vaporizes liquid nitrogen under pressure and produces compressed nitrogen gas. This compressed gas runs a pneumatic (compressed-gas-driven) motor with nitrogen gas as the exhaust.

What are the components of this engine?
There are three principal components:

- a pressurized tank to store the liquid nitrogen
- a heat exchanger that heats (using atmospheric heat) liquid nitrogen to form nitrogen gas, then heats the nitrogen gas under pressure to near atmospheric temperature
- a pneumatic motor (along with a Volkswagen transmission) that runs the car

This vehicle was assembled using "off-the-shelf" components.

What's it like to drive the CooLN2Car? How far can it go and how fast?
The CooLN2Car can travel 15 miles on a full (48 gallon) tank of liquid nitrogen going 20 mph. Its maximum speed is over 35 mph. In general, the driving characteristics of the CooLN2Car are very similar to those of a regular car. Other than the limited speed, the most noticeable difference is that the motor does not idle while the vehicle is stopped.

How would electrical elements be powered in a nitrogen-powered car?
Alternators, generators, photovoltaics and thermoelectric materials all could effectively produce sufficient electricity to run auxiliary devices.

Does this vehicle pollute?
No, the exhaust gas is nitrogen, the major constituent of our air. In fact, liquefied air could also be used to power this vehicle.

How would enough liquid nitrogen or liquefied air be produced to power a fleet of cryogenic heat engine cars?
Off-peak power from nuclear power plants could produce enough energy to condense nitrogen or air. Because liquid nitrogen production requires additional distillation steps, liquefied air would be less expensive to produce. We are examining ways to use solar powered gas liquefaction technologies.

Is there a market for these vehicles?
By 2003, 10% of new automobiles sold in California must be zero emission vehicles. Our calculations

indicate that a cryogenic heat engine car could meet or exceed the range and performance of a battery powered car; therefore, nitrogen powered vehicles could compete for a share of the zero emission market in 2003.

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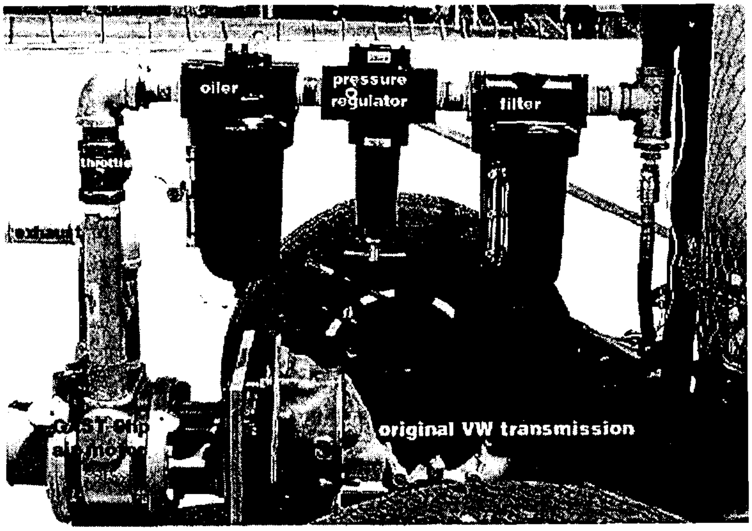
POWER TRAIN OF NITROGEN-POWERED CAR



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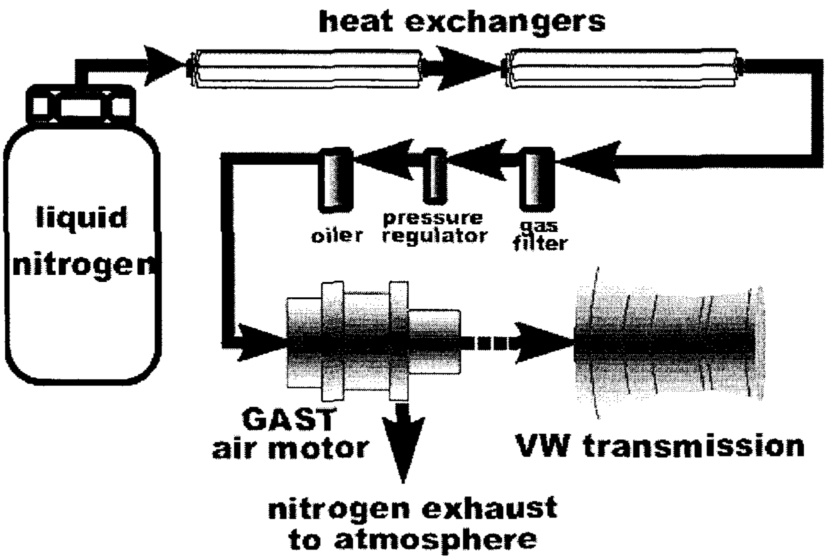
GAS FLOW DIAGRAM FOR OUR NITROGEN-POWERED CAR



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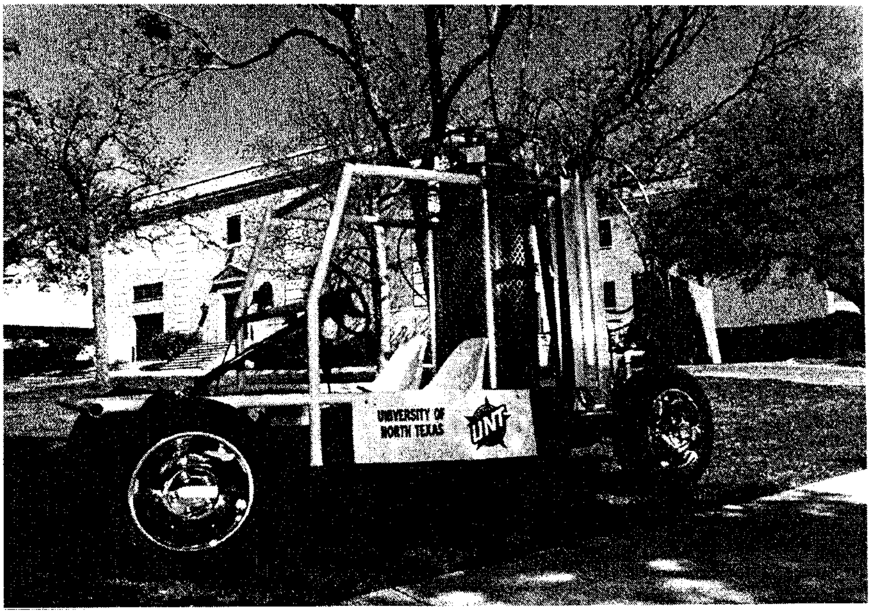
Website maintained by R.F. Reidy and last updated August 6, 1997.

A Car Powered by Liquid Nitrogen!



University of North Texas CooLN2Car

Designed, built, and tested by Dr. Carlos Ordonez (Physics), Dr. Mitty Plummer (Engineering Technology), and Dr. Rick Reidy (Department of Materials Science) of the University of North Texas . this developmental zero emission vehicle employs a cryogenic heat engine and is fueled by liquid nitrogen. This research was funded by the Texas Advanced Technology Program. Details of this work were presented in July 1997 at the Cryogenic Materials Conference in Portland, Oregon. Questions about this program can be addressed to:

Dr. Carlos Ordonez
Department of Physics
PO Box 311427
University of North Texas
Denton, Texas 76203-1427
940-565-4860

Dr. Mitty Plummer
Dept. of Engineering Technology
PO Box 13198
University of North Texas
Denton, Texas 76203
940-565-2846
email: plummer@unt.edu

Dr. Rick Reidy
Dept. of Materials Science
PO Box 305310
University of North Texas
Denton, Texas 76203-5310
940-369-7115
email: reidy@unt.edu

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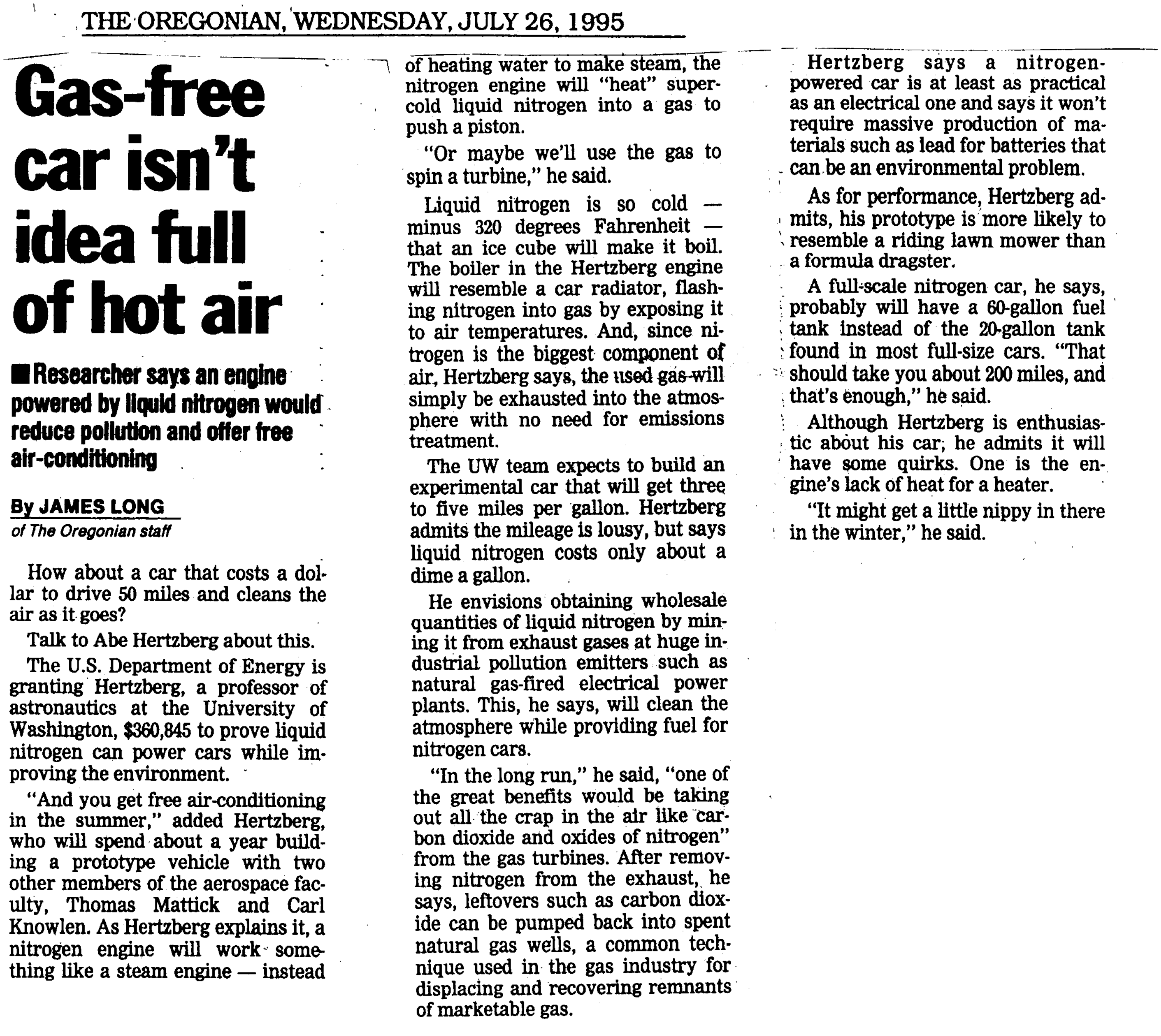
A University of Washington group has a similar liquid nitrogen car program.

You may wish to visit their website at http://www.aa.washington.edu/AERP/CRYOCAR/.

ADDITIONAL INFORMATION:

Our August 8th press release Our Papers

Website maintained by R.F. Reidy and last updated December 20, 2001.



THE OREGONIAN, WEDNESDAY, JULY 26, 1995

Gas-free car isn't idea full of hot air

■ Researcher says an engine powered by liquid nitrogen would reduce pollution and offer free air-conditioning

By JAMES LONG
of The Oregonian staff

How about a car that costs a dollar to drive 50 miles and cleans the air as it goes?

Talk to Abe Hertzberg about this.

The U.S. Department of Energy is granting Hertzberg, a professor of astronautics at the University of Washington, $360,845 to prove liquid nitrogen can power cars while improving the environment.

"And you get free air-conditioning in the summer," added Hertzberg, who will spend about a year building a prototype vehicle with two other members of the aerospace faculty, Thomas Mattick and Carl Knowlen. As Hertzberg explains it, a nitrogen engine will work something like a steam engine — instead of heating water to make steam, the nitrogen engine will "heat" super-cold liquid nitrogen into a gas to push a piston.

"Or maybe we'll use the gas to spin a turbine," he said.

Liquid nitrogen is so cold — minus 320 degrees Fahrenheit — that an ice cube will make it boil. The boiler in the Hertzberg engine will resemble a car radiator, flashing nitrogen into gas by exposing it to air temperatures. And, since nitrogen is the biggest component of air, Hertzberg says, the used gas will simply be exhausted into the atmosphere with no need for emissions treatment.

The UW team expects to build an experimental car that will get three to five miles per gallon. Hertzberg admits the mileage is lousy, but says liquid nitrogen costs only about a dime a gallon.

He envisions obtaining wholesale quantities of liquid nitrogen by mining it from exhaust gases at huge industrial pollution emitters such as natural gas-fired electrical power plants. This, he says, will clean the atmosphere while providing fuel for nitrogen cars.

"In the long run," he said, "one of the great benefits would be taking out all the crap in the air like carbon dioxide and oxides of nitrogen" from the gas turbines. After removing nitrogen from the exhaust, he says, leftovers such as carbon dioxide can be pumped back into spent natural gas wells, a common technique used in the gas industry for displacing and recovering remnants of marketable gas.

Hertzberg says a nitrogen-powered car is at least as practical as an electrical one and says it won't require massive production of materials such as lead for batteries that can be an environmental problem.

As for performance, Hertzberg admits, his prototype is more likely to resemble a riding lawn mower than a formula dragster.

A full-scale nitrogen car, he says, probably will have a 60-gallon fuel tank instead of the 20-gallon tank found in most full-size cars. "That should take you about 200 miles, and that's enough," he said.

Although Hertzberg is enthusiastic about his car, he admits it will have some quirks. One is the engine's lack of heat for a heater.

"It might get a little nippy in there in the winter," he said.

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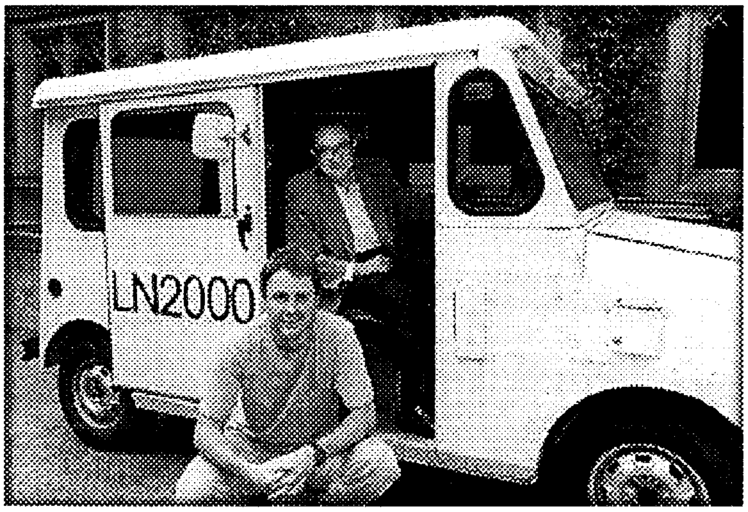
This Car Runs on Air



Liquid Nitrogen Fuel

Abe Hertzberg, professor emeritus of aeronautics and astronautics at the University of Washington, and one of his students pose with the LN2000, the nitrogen-powered vehicle they have developed (Universtiy of Washington)

SCIENCE HEADLINES

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Special to ABCNEWS.com

LEE DYE

How about a car that runs on air, doesn't poison the atmosphere and actually helps clean up air pollution?

Sounds too good to be true, but that's what researchers at the University of Washington are trying to develop. Expanding on a concept that has been tried before and abandoned, the researchers have even built a crude prototype of a car that looks like an old mail truck, which is exactly what it is.

It is all the brainchild of Abe Hertzberg, professor emeritus of aeronautics and astronautics at the university, who heads up "project smogmobile," named after the car in the L'il Abner cartoon that runs on air pollution.

The vehicle runs on nitrogen, which makes up 78 percent of our atmosphere.

Nitro, Naturally

The contraption built by Hertzberg and researchers and students at the university doesn't quite do that, but it comes pretty close. It runs on nitrogen, which makes up 78 percent of our atmosphere. Hertzberg believes it could be a viable substitute to gasoline-powered vehicles of today and the electric-powered cars now being mandated by some states, especially California.

Funded by a $360,000 grant from the U.S. Department of Energy, the researchers took an ancient "air motor" and converted it to run on nitrogen.

They mounted it in an old mail truck and "it ran the first time we took it out of the shop," Hertzberg says. "About 25 people have already ridden around the block a few times."

How it Works

The underlying concept behind the research was to keep it simple, he adds. Here is how it works:

Nitrogen gas is extracted from the atmosphere and supercooled to -320 degrees Fahrenheit, which converts it to a liquid. The liquid nitrogen is then pumped through a heat exchanger, like the radiator in a conventional car, which uses heat already present in the atmosphere to warm it up enough for the liquid to vaporize.

As it converts from liquid to gas, the nitrogen expands to 700 times the volume of its liquid form. The expanding gas pushes on a piston in the "air motor," forcing it to move.

That is similar to the process that drives an internal combustion engine—gas vapors are ignited and expand rapidly, pushing a piston down and turning a crankshaft that propels the vehicle To test their concept, the researchers needed to find an "air motor" that used compressed air to drive the pistons. The best they could come up with was a relic from the past—a 1941 anchor winch that weighed 600 pounds and was in such poor condition that it had to be rebuilt.

It eventually worked, but only produced about 15 horsepower. That was enough to verify the concept, and Hertzberg's team hopes to improve on the performance in subsequent versions of their "smogmobile."

Similar efforts in the past have failed, Hertzberg says, because frost built up in the heat exchanger, stalling the process. His team claims to be the first to develop a heat exchanger that doesn't freeze up.

Better Than Electric Cars

Hertzberg hopes his project will help curb enthusiasm for electric cars, which he says are inefficient, inconvenient, and not nearly as environmentally-friendly as they are alleged to be. Batteries have to be recharged, for example, increasing the demands on power plants, and they use toxic metals that have to be carefully disposed of.

"If you spill liquid nitrogen, it just sits there in a pool and slowly evaporates" back into the atmosphere from where it came, he says.

What particularly excites him is the potential to help clean up the atmosphere. Nitrogen can be extracted from the air and liquified in giant refrigeration plants, he says. In the same process, pollutants such as carbon dioxide could be removed from the air. And he claims it could all be done cheaper than it costs to produce a comparable amount of gasoline.

If he has his way about it, one of these days you will be able to pull in to your neighborhood gas station and fill up with about 100 gallons liquid nitrogen, enough to drive about 250 miles. But he says he isn't going to hold his breath.

The automotive industry, he says, "looks at a new idea the same way you and I would look at a rattlesnake."

It would require very fundamental changes in the automotive infrastructure, including massive distribution systems for liquid nitrogen. Even Hertzberg admits he doesn't see that happening.

So what does he expect to get out of it?

"I expect to get my head beat in," he says.

But he hopes that, in the end, a few more people will agree with him that gasoline and electricity are poor ways to run an automobile. Something more benign that uses an inexhaustible fuel—like nitrogen—must be found.

Science writer Lee Dye is a regular contributor to ABCNEWS.com

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INTERVIEW WITH PAUL TRENTHAM, CONSULTING ENGINEER
By Scott Robertson, Pneumatic Options Research Library
www.AirCarAccess.com

RE: Paul Trentham, PE
7342 Old Scenic Dr.
Joplin, MO 64801
417-623-5459
tigertrentham@4state.com

4/14/04

I called Paul Trentham, a consulting engineer in Joplin, Missouri because I had his name in connection with the Terry Miller/Toby Butterfield air car project from the previous decade. I wanted to find out if he knew the fate of the project, and especially what happened to the three air cars I knew of that Terry had built. The interview was not recorded, but I have summarized it below based on notes from our conversation.

Paul Trentham is a consulting engineer, among whose accomplishments is that he worked on the Apollo engines for the moon shot in the 1960s.

Paul got involved with the Joplin air car project about the time Terry got out of it. (Editor's note: Terry was the inventor, and Toby his business partner; Terry gave his share to his daughter, Teresa Robbins. Terry and Toby are now deceased.) Toby called Paul and explained that he and Terry had come to an impasse, that the partnership was not working out, and that Terry had lost interest in pursuing it. Paul was asked to make an analysis of the invention to determine if there was any practical use in continuing it. Paul only met Terry once, and was not well acquainted with Ms. Robbins, Terry's daughter. His dealings were with Toby.

Regarding the air cars that Terry built, Paul recalls only the blue Chevy Sprint (editor's note: there was also the famed "Air Car One," a three-wheeled workbench on wheels that Terry showed from coast to coast (1980s), and a red compact car, probably either a Chevy Sprint or a Ford Festiva (1990s, called "Miss Rexroth," the blue car was the "Spirit of Joplin")). Paul met Teresa Robbins, Terry's daughter, at a meeting where the project collapsed and everyone went their separate ways. Ms. Robbins has moved away from the Joplin area. The project had no tangible assets at the time of its dissolution, and the blue car was mashed for scrap. Paul never saw Air Car One. He still has a number of high pressure air tanks, originally made for CNG usage, and would like to sell them. Most have never been used.

Paul wrote his report in about 1992 or 1993. The good news was that he found a definite use for compressed air as a motive power for vehicles. The bad news was that he felt Terry's way of going about it was all wrong. In Paul's opinion, which was based on computer models and other research, multiple expansion is impractical because it can only be optimized at one set of conditions: one speed, one cutoff. He is convinced that the 2nd, 3rd, and 4th expansion stages on Terry's engine were actually dragging on the 1st stage, instead of augmenting its power. Paul tested the car with a simple Prony Brake placed on a front driving wheel, the other driving wheel on the ground. The power of the engine was measured and found to be pulsating and unsteady, falling to zero and rising to about 1.5 hp. This seemed reasonable with the knowledge that all of Terry's cars needed pushing to get them started, even on a flat surface. He felt that compound engines could work well for a steady speed stationary engine, but less so for locomotion, and certainly not more than two stages total. He cited Norfolk and Western Railroad efforts with two stage compound steam locomotives based on the Mallet design.originating in France. The compound idea was well known in The Steam Age, but never more than two stages in locomotives, and for good reason.

He felt it would be much more practical, in a vehicle engine, to use a single-stage engine with variable cut off. Some air is used inefficiently at long cut offs required for acceleration from zero, but the time is so short the loss is not worth the extra complexity of multiple compounding. The most economical use of air seems to be at a very short cut off of only 5%. To realize high efficiency at such a short cutoff requires near zero clearance space in the engine cylinders, and very good seals. Rotary valves help to reduce unwanted clearance space, and have no reciprocating mass problems causing vibration and wasted power loss.

Toby asked Paul to design a better engine, which he did. He determined that the most efficient cutoff point would be 5%, but in order to start and accelerate the car, something like 50-60% would be needed. At this cutoff range, the engine would develop enormous torque, as in steam engines, and no transmission would be needed. (Note that Terry's vehicles also had no transmission, but never achieved much torque or power due to the ineffective and complicated multiple compounding scheme.)

Paul designed and built a variable cutoff engine over the next three years, converting a Honda 500cc V-twin motorcycle engine to run on air. He moved the equipment from the Joplin air station to his home and built a garage and shop to house the project. He invented a rotary valve to eliminate complicated reciprocating valve parts, which allowed faster engine speeds than the old-fashioned expansion engines were capable of, and a simple cut off control valve. One rotary valve took care of intake and exhaust, (flow to and from the cylinder) while another small variable cut off valve ahead of the intake/exhaust valve adjusted the cutoff (the amount of stoke during which the inlet valve was open.) Although he was not happy with the seals on the valves, it did work. It developed about 9.5 horsepower at 5% cutoff, and was operated up to 800 psi. Higher pressures were not used due to the leaky valve seals.

He accidentally discovered that the engine would function well at up to 14,700 rpm, by opening the engine full throttle from rest, with no load. The engine jumped almost immediately to that speed, and ran at that speed with almost no vibration. The compressed air was entering the engine at the pressure tank's ambient temperature of 83° F., but he had trouble measuring the exhaust temperature at 5% cutoff because the expanded air bottomed out his temperature gauge, which couldn't go below -75°. He has a VHS tape somewhere, showing that if you blew across the engine's exhaust stream, the moisture in your breath would freeze and fall to the ground as snow. The engine ran smoothly down to about 30 rpm.

The University of Washington, Seattle, got interested in the engine. The Aerospace Engineering Department had been working on a liquid nitrogen car, which used a marine auxiliary steam engine they had found and adapted for their purposes by installing it to run a small mail truck. They invented and patented a triple-pass heat exchanger that warmed liquid nitrogen up to nearly ambient temperature, using the free energy of the atmosphere to do the warming. Paul spoke very highly of their heat exchanger, saying that it did not freeze up and functioned very well. After trying the former steam engine, they had some grant money left, so Paul sold the engine he had developed to the U of W and they put it in their cryogenic car. The young professor working on the project was Dr. Carl Knowlen, whose address is U of W College of Engineering, Aerospace Building Room 136, Seattle, WA. The head of the department is Dr. Adam Bruckner, and his predecessor, Dr. Abe Hertzberg was the founder of the nitrogen car project. Dr. Knowlen wrote SAE papers on the car, but the project ran out of grants.

Paul questioned the use of liquid nitrogen by the U of W project, rather than liquid air. The U of W was concerned about using liquid air because they thought the oxygen would separate and cause a fire hazard in an accident. Paul says that since the molecular weights of oxygen and nitrogen are very close, the cost of a stirring device to keep them from separating in storage would be insignificant. Dr. Knowlen studied the use of cryogenic air and nitrogen and the economics and safety issues, and delivered a paper at the Society of Automobile Engineers conference in Los Angeles. This was during the popularity of battery powered cars, so it did not get a lot of attention.

Paul still thinks that air is the way to go. He never got to finish what he wanted to do with his engine. He is now working on a rotary valve for internal combustion engines, with fully variable timing. He has been granted two patents for his rotary valve and has another patent in the works.

The seals on the original short cutoff engine limited the pressure and therefore the effectiveness and efficiency of that engine. A different approach on the seals would be used if further air engine designs were undertaken.

Paul notes that a cryogenic car can be run on ordinary compressed air by bypassing the heat exchanger, although the inverse is not true. He states that the cryogenic car has the advantage over the compressed air car, in that it can store 2.5 times more air in a given tank at a given pressure. Therefore the cryogenic car can exceed the range of the pneumatic car by more than twice. The cost of doing that is to have a U of W type heat exchanger on board to warm the cryogenic air up to near ambient, with the "free" ambient energy. It is this ambient energy converted back to useful driving energy that extends the driving range of the car by double over the compressed air concept. Another way of stating it is that the car can carry twice as much mass of air as a liquid, but it all comes out to the same impressive gain in on board energy storage. You get air conditioning as a side benefit whether the car is running on cryogenic air or highly pressurized air.

Paul is still interested in working on air engines and would be willing to be hired on to do engineering work on either cryogenic or pneumatic systems. He strongly believes in the superiority of the cryogenic system, which would need to be supported by tanker trucks transporting liquid air to cryogenic air stations, because of the greater range between fill-ups compared to the compressed air car.

Paul's own written explanation of the advantage of cryogenic-over-compressed air is reproduced here:

> "Liquid air is much denser than pressurized air. We were using lightweight tanks designed for and used by the CNG (compressed natural gas) cars. CNG burns cleaner than gasoline, and is used in some places, but has much lower energy availability, generates Carbon Dioxide, etc., so it probably won't ever be THE solution. These tanks are good for, I believe, 4000 psi. Because cryogenic air (liquid air) is a liquid, instead of a gas, it is much denser, or about 2.5 times as heavy as even the highly pressurized air at around 4,000 psi. (I hope my numbers are right, it has been several years since I went through this). You see, air is quite light at atmospheric pressure, but it gets more dense as you pressurize it. If you pressurize it to two atmospheres, that is to say, about 30 psi instead of 14.7 psi, then it is twice as dense, because you have twice as many molecules in the same volume. And so forth. And when you go from high pressure (about 3500 or 4,000 psi) to liquid, you get about 2.5 times as much liquid air in the same volume tank that held the high pressure air. Above 4,000 psi, it is not practical to pressurize air and haul it around in a tank, it is generally agreed it is too dangerous if ruptured, it is like a damn big gun exploding. So that is sort of the agreed on upper limit for hauling around pressure vessels of anything safely on the highways, air included. Plus the tanks get to be very heavy walled, and expensive, and weigh a lot.

“Liquid air is made in large plants by pressurizing the air, and then removing the heat caused from the pressurization, and then doing it again and again, until it collapses from the gaseous state to a liquid, and it is very cold, I can't remember, lousy memory, but it is around 200 degrees F below zero. Now it is not under great pressure as a liquid, but will vaporize and turn back to gaseous air as ambient heat leaks in through the tank, and that causes the pressure to rise, so you have to insulate the container, and that is often done by making a Dewar flask, which is like a thermos bottle. Fortunately there are some really good insulators from the past few years, and you can insulate a tank with stuff like glass foams that do a great job of keeping the liquid air cold. Even so, there is a little heat leaks in, and I think they figure on losing about 5% in a week, due to heat leaks, and having to vent the excess pressure off.

”Dr. Knowlen and I looked at this several years ago and he wrote a paper on it, so you guys might call Washington U in Seattle and ask him for a copy. I got one somewhere, but it is filed away in the junk, but I remember that we found out from the big suppliers that it costs about 15 cents a gallon to make liquid air. Carl figured it out for reasonable mileage and storage, and I think it came out that liquid air was the equivalent, economically, of operating a gasoline powered car at $1.30 per gallon. So you see, the economics are not at all bad, and no gas price wars with the Arabs and Californians driven by environmentalists to not build more refineries, etc.

”Liquid cryogens, mostly Oxygen, Nitrogen, Argon, and CO2, are hauled all over the country in those long shiny tank tractor trailers, and I have never heard of a bad accident like you have with gasoline tankers. The big liquid air plants are expensive to build, so they haul their product hundreds of miles. They crack off Oxygen, Nitrogen, Argon, etc. from these plants much like you do in a petroleum refinery, based on their liquification temperature. They don't explode, not even Oxygen, because there is no fuel to ignite when the tank breaks. Remember, the pressure is not high!!! Oh the oxygen will burn anything organic it touches, like gasoline, that is how the Moon rockets worked, and you don't need a spark plug. But by and large it is safer to store liquid air than it is to store gasoline, in my opinion.

”You could have filling stations just like you have now, only they would have insulated liquid air tanks. They get filled once a week by the big highway tankers. Special nozzles used to fill the liquid air tank in your car, not a big deal, and not at high pressure like the pressurized air tanks. If you have a leak, so what? The liquid air quickly vaporizes, and turns into air, and it is much cleaner than the air in LA, because liquifying it takes out all the dirty stuff. Neat, hunh?

"Still won't have as good mileage range as gasoline powered cars, but about double what you will have in pressurized air cars. And if you are remotely located and don't have a liquified air station, just use your handy dandy high pressure air compressor and run the car at half its liquid air range. The liquid air tank will probably have the same pressure rating as the high pressure air tank.

"A word about air conditioning. You don't get the AC from the very cold liquid air, you get it from the expansion of the air in the cylinder. So you get A/C whether the car is liquid air or pressurized air tank. I was taking air at about 80 degrees F from a pressurized tank and putting it through my engine at several hundred psi, and it was expanding to do work in the cylinder and coming out the exhaust at below 75 degrees below zero and you could make it snow by blowing your moisture laden breath across the exhaust. My seals were not very god, so I only ran at 300 to 700 psi, and that is not very efficient use of the air, but the engine ran as high as 14,700 rpm, and, by the way, there was almost NO vibration, and it would run nicely down to about 30 rpm. You do not need much if any transmission in a "steam" (Rankine thermodynamic cycle) engine. Lots of torque when you need it for short time, though inefficient at long cut offs.

"Cryogenic air doubles your tank range, and it is safer than high pressure air. You can use the same vehicle and tank for high pressure air, it is just a little bigger with the insulation. I would rather hit a cryogenic tank full of liquid air, than a 4,000 psi air tank. The difference in explosive power and potential hazard in an accident is enormous. Spilled cryogenic air on asphalt, might or might not burn the asphalt, I don't think so, as 80 percent of the liquid air is Nitrogen, and non flammable. It boils off to become air very rapidly. That is why you want to use liquid air, rather than liquid oxygen, and you don't need to go all the way to liquid Nitrogen to make it safe, but some tests would be needed."

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C. Knowlen, A. Hertzberg, and A.T. Mattick
Aerospace and Energetics Research Program
University of Washington
Seattle, WA 98195

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CRYOGENIC AUTOMOTIVE PROPULSION

C. Knowlen,* A. Hertzberg,† and A.T. Mattick‡
Aerospace and Energetics Research Program
University of Washington, FL-10
Seattle, WA 98195

ABSTRACT

Cryogens are effective thermal storage media which, when used for automotive purposes, offer significant advantages over current and proposed electrochemical battery technologies, both in performance and economy. An automotive propulsion concept is presented which utilizes liquid nitrogen as the working fluid for an open Rankine cycle. When the only heat input to the engine is supplied by ambient heat exchangers, an automobile can readily be propelled while satisfying stringent tailpipe emission standards. Nitrogen propulsive systems can provide automotive ranges of nearly 400 kilometers in the zero emission mode, with lower operating costs than those of the electric vehicles currently being considered for mass production. In geographical regions that allow ultra low emission vehicles, the range and performance of the liquid nitrogen automobile can be significantly extended by the addition of a small efficient burner. Some of the advantages of a transportation infrastructure based on liquid nitrogen are that recharging the energy storage system only requires minutes and there are minimal environmental hazards associated with the manufacture and utilization of the cryogenic "fuel."

I. INTRODUCTION

The development of road-worthy electric vehicles (EVs) has been stimulated by the approach California has taken to improve its urban air quality to meet Federal standards. In 1990 the state government passed legislation requiring that, starting in 1998, at least 2% of all new vehicles sold within its jurisdiction to be zero emission vehicles (ZEVs), with the minimum fraction of new vehicle sales to increase to 10% by the year 2003. Several other states have adopted the California regulations and full compliance would result in over 500,000 new ZEVs being sold every year in just the U.S.A. The environmental impact of the energy storage technology chosen to power such a large fleet of vehicles must be carefully evaluated.

Under present regulations, a ZEV is one that does not produce any tailpipe pollutants, regardless of the emissions produced in the manufacture of the vehicle or in generating the electricity to recharge its energy storage system. Some of the available technologies for storing energy that meet these qualifications are electrochemical batteries, fuel cells, and flywheels.[1] Improvements in each of these energy storage systems continue to be made; however, only the electrochemical battery has reached a high enough state of development to be considered useful and practical in a large EV fleet. Among the different battery concepts being developed, the lead-acid battery still appears to offer the best compromise of performance, utility, and economy for transportation applications.

A significant fraction of the lead produced each year is used in lead-acid batteries for automobiles and a typical EV requires 20-30 conventional batteries to have a useful range. The growing public awareness of the health hazards arising from elevated concentrations of lead in the environment has resulted in a steady decrease in the amounts of lead used in industry and personal products over the years. In addition, it has been found that the pollution control practices at the mine heads and ore smelters have not prevented serious degradation of their surroundings. Thus, it is highly probable that the environmental impact of the increased mining and refining of lead ore, to meet the needs of a transportation infrastructure based on lead-acid batteries, could completely negate the benefits expected from the elimination of tailpipe emissions.[2]

Due to the expected degradation of the environment arising from the wide spread use of lead-acid batteries and the technical difficulties of applying other advanced energy storage technologies, we have been examining the potential of using cryogens as high energy density thermal storage media. If the cryogen can be effectively used in a heat engine whose only energy input comes from the atmosphere, then a new kind of EV is possible which avoids many of the environmental consequences of electrochemical batteries. Indeed, it appears that liquid nitrogen (LN_2) can be used in a zero-emission propulsion system that is as effective and probably more economical to

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* Research Associate, Member AIAA
† Professor Emeritus, Fellow AIAA
‡ Associate Professor, Member AIAA

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operate than the high performance battery systems currently under development.[3]

There have been several concepts for storing the working fluid of a propulsion system as a cryogenic liquid or frozen solid discussed in the open literature. One of the earliest articles describes the use of dry ice to power the steam locomotive of a model train set.[4] Sublimation of the CO_2 rapidly pressurized the tender reservoir which enabled operation for up to an hour. Even if this approach was practical to scale up, there would be significant opposition to the wide spread release of such a major greenhouse gas. Others have suggested using methane as the working fluid of a Rankine cycle operating on the temperature difference between a liquid hydrogen reservoir and ambient conditions.[5] A simple prototype was built which generated approximately 1 KW with 26% thermal efficiency in laboratory tests using LN_2 in place of liquid hydrogen. This device was successful in demonstrating a means for driving the pump needed to pressurize the fuel system of a hydrogen-powered automobile without sacrificing any engine power.

Around 1980, a full-sized automobile propulsion system using a diesel-fired combustor to vaporize and superheat N_2 to approximately 1000 K was suggested as a means to reduce our dependence on imported oil.[6,7] This alternative power plant concept appeared capable of providing the same road performance as a conventional automobile while having comparable operating expenses and a much lower fuel consumption. Apparently the economics of introducing this technology to compete with gasoline engines were not favorable enough to warrant further development at that time. In any case, since there is ample enthalpy in ambient air to vaporize LN_2 with only a few degrees of temperature drop at low air flow rates, the use of a high temperature combustor is not necessary to adequately power small commuter vehicles.

The LN_2 propulsion system for a ZEV operates very much like a conventional steam engine while taking advantage of machinery designed for subambient temperature applications. Instead of using a steam jacket to minimize the heat loss through the steam cylinder walls, a LN_2 expander will have circulating fluid to maintain the wall temperature as high as possible to enhance heat transfer during the power stroke. A schematic of a Rankine cycle propulsion system using only ambient air to heat the cryogenic working fluid is shown in Fig. 1. An insulated "fuel" tank contains LN_2 at atmospheric pressure and a temperature of 77 K. A cryogenic pump draws LN_2 from the bottom of the tank and compresses it to the operating pressure of the system.

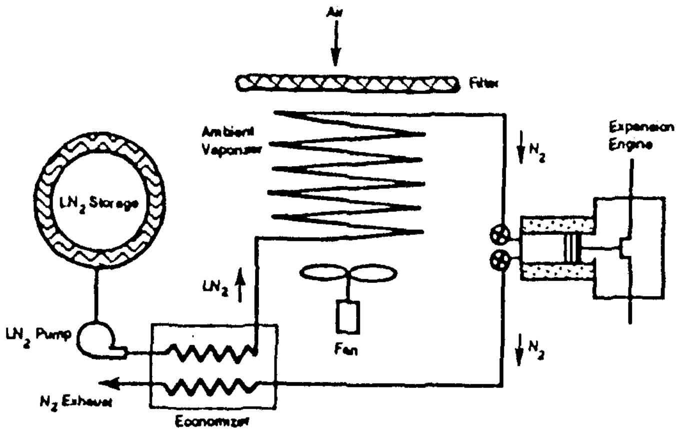

Fig. 1 Schematic of LN_2 propulsion system using only ambient air for heat source.

Pressurized LN_2 then passes through the heat exchanger, which is optimized as a LN_2 vaporizer and N_2 superheater, to raise the gas temperature to just below ambient conditions. The gaseous N_2 is then injected into the cylinder as the piston approaches top dead center. It is possible for multiple expansion strokes and reheats to be utilized, at the expense of mechanical complexity, to approach quasi-isothermal performance. If the exhaust gas has sufficient enthalpy it passes through an economizer before being vented to atmosphere. The nitrogen condensation phase closing the Rankine cycle occurs at stationary air liquefaction plants.

This zero emission propulsion concept offers many environmental advantages over internal combustion engines and electrochemical battery vehicles. It has low operating costs, ample propulsive power, and reasonable round trip energy efficiency. We refer to this ZEV as the "cryomobile." In the following section the thermodynamics of cryogen energy storage are presented, with the primary emphasis on liquid nitrogen. An application of LN_2 to pollution-free automotive propulsion is then discussed along with some of the technical issues facing its implementation and the possibility of range extension using a small burner.

II. THERMODYNAMICS OF CRYOGENS

The basic idea of the LN_2 propulsion system is to utilize the atmosphere as a heat source and a cryogen as a heat sink in a thermal power cycle. This is in contrast to typical thermal engines which utilize an energy source at temperature significantly above ambient and use the atmosphere as a heat sink. In both cases the efficiency of conversion of thermal energy of the source to work (W) is limited by the Carnot efficiency, $\eta = W/Q_h = 1 - T_l/T_h$, where Q_h is heat input, T_l is the sink temperature, and T_h is the temperature of the heat source. By using liquid nitrogen as the cryomobile energy sink ($T_l = 77$ K) this ideal thermal efficiency is impressively high (74%) with an atmospheric heat source at $T_h = 300$ K. The key issues are the ability to design a practical energy conversion system that can take advantage of this high efficiency and the available energy of the cryogen while still being cost competitive with alternative EVs.

Thermal Storage Effectiveness

Because the design and expense of the cryomobile are driven primarily by the cryogen heat sink, it is useful to characterize performance by a "sink efficiency", $\beta = W/Q_l$, i.e. the inverse of the coefficient of performance for a refrigerator. For a reversible cycle with fixed sink and source temperatures the ideal sink efficiency is $\beta = T_h/T_l - 1$, or 2.9 for $T_l = 77$ K and $T_h = 300$ K. If only the enthalpy of evaporation of LN_2 ($h_{fg} = 199$ kJ/kg) is used to sink such an engine, the ideal work output is 576 kJ/kg of LN_2. This specific energy is significantly higher than the 180-300 kJ/kg achieved with lead-acid batteries.[8]

Additional potential remains for heat sinking by the cold N_2 vapor, and the ideal work recoverable from an expansion engine as LN_2 is evaporated at 77 K, then brought up to ambient temperature ($T_o = 300$ K), is given by the difference in thermodynamic availability, $\Psi = h - sT_o$, between liquid and ambient states. This "reversible work" for LN_2 is $W_r = 769$ kJ/kg-N_2 and the corresponding sink efficiency is $\beta = 1.78$. Thus, only a small fraction of the reversible work needs to be recovered to provide the cryomobile with a driving range commensurate with that of battery-powered vehicles of comparable weight. The relative merits of different ZEV technologies must also be evaluated on considerations other than performance such as environmental friendliness and commuting utility. While the specific energy of LN_2 or batteries is far below that of hydrocarbon fuels, the internal combustion engine cars cannot operate without polluting emissions and thus should not be compared with ZEVs.

Power Cycle

The Rankine cycle is among the most attractive choices for approximating Carnot performance, when using a fixed temperature heat source and sink. We have focused on directly using the nitrogen itself as the working fluid, wherein the liquid is compressed with a cryogen pump, heated and vaporized by heat exchange with the atmosphere, and then expanded in a piston-cylinder engine. The temperature-entropy diagram for a representative case is illustrated in Fig. 2. State 1 is the cryogenic liquid in storage at 0.1 MPa and 77 K. The liquid is pumped up to system pressure of 4 MPa (supercritical) at state 2 and then enters the economizer. State 3 indicates N_2 properties after its been preheated by the exhaust gas. Further heat exchange with ambient air brings the N_2 to 300 K at state 4, ready for expansion. Isothermal expansion to 0.11 MPa at state 5 would result in the N_2 exhaust having enough enthalpy to heat the LN_2 to above its critical temperature in the economizer, whereas adiabatic expansion to state 6 would not leave sufficient enthalpy to justify its use. The specific work output would be 320 and 200 kJ/kg-LN_2 for these isothermal and adiabatic cycles, respectively, without considering pump work. While these power cycles do not make best use of the thermodynamic potential of the LN_2, they do provide specific energies competitive with those of lead-acid batteries.

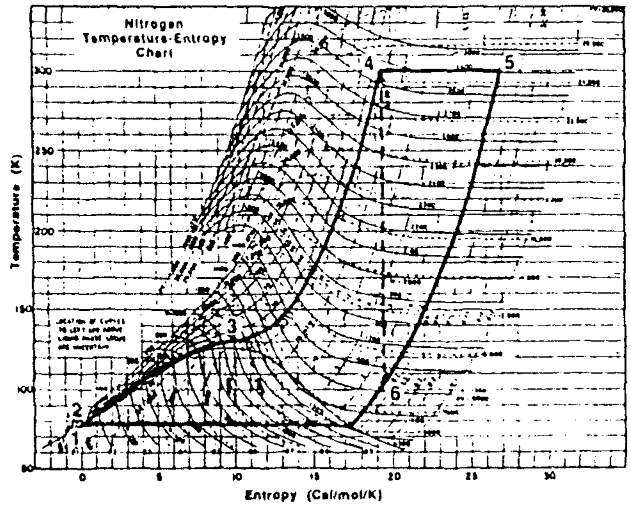

Fig. 2 Temperature-entropy diagram of Rankine cycle using LN_2 for working fluid.

Liquid Nitrogen Manufacture

The cost of the LN_2 "fuel" is expected to be reasonable. The primary expense for producing LN_2 is the energy cost for compression of air.[9] Cryogenic separation of nitrogen from other condensables in air typically requires only a very small fraction of the total energy,[10] so the ideal work to manufacture LN_2

from air is very nearly that for using nitrogen as a feedstock. This work is exactly the reversible work obtainable from an ideal cryo-engine, 769 kJ/kg. The actual work required in a modern LN_2 plant is 2.0-2.5 times the minimum, or 1540-1920 kJ/kg.[11] Assuming an industrial electric rate for interruptible power of 5¢/kW-h, the energy cost would amount to 2.6¢/kg-LN_2, in accord with delivery prices of LN_2 in large quantities. Marketing the other commercially important components of air will help offset the LN_2 production costs. Since the equipment needed for air liquefaction can be powered solely by electricity, it is conceivable to decentralize the "fuel" manufacturing process and to place small scale production facilities at the LN_2 dispensing sites. A cost-benefit analysis is needed to determine the smallest air liquefaction machinery that can be used to produce LN_2 in an economical manner.

III. APPLICATION TO AUTOMOTIVE PROPULSION

In order to determine the effectiveness of using LN_2 as the working fluid, the performance capabilities of reciprocating engines having adiabatic or isothermal power strokes (infinite reheats) have been examined. Curves indicating the specific work for both expansion processes as a function of injection pressure are shown in Fig. 3, with the injection temperature as a parameter. The LN_2 pump is assumed to compress the cryogenic liquid to operating pressure with 80% efficiency. An ideal expander is assumed and the LN_2 injection cut-off point has been adjusted so that the final pressure after expansion is always 0.11 MPa. For the range of pressures shown in Fig. 3, the work of the isothermal cycle increases monotonically with increasing pressure, whereas the adiabatic LN_2 engine is only weakly dependent on peak cycle pressure above 4 MPa. Since this is a reasonable operating pressure for both piston and turbine expanders, it will be used to illustrate the performance potential of the LN_2 energy storage system.

The cruising characteristics of a cryomobile, operating with either the adiabatic or isothermal cycle, are presented in Table 1. A peak injection pressure of 4 MPa and peak cycle temperature of 300 K are used here. Estimates of the mass and volume of the LN_2 required for a given range are based on a vehicle that requires the same amount of road power during freeway cruise as the Honda CRX automobile, i.e., 7.8 kW at 97 km/h (60 mi/h).[12] The "fuel" operating costs are based on the economics of supplying LN_2 from a plant optimized for its production (2.6¢ per kg-LN_2), as discussed in Section II. For the specified cruise conditions, the propulsion system having an isothermal expansion process will consume LN_2 at a rate of 25 gm/sec and have an operating cost of approximately 2.4¢ per kilometer. The corresponding storage volume and mass of LN_2 required to provide a maximum range of 300 km is 400 liters (106 gal) and 280 kg, respectively. Thus the cryogenic storage tank can readily fit within the trunk volume of a conventional automobile.

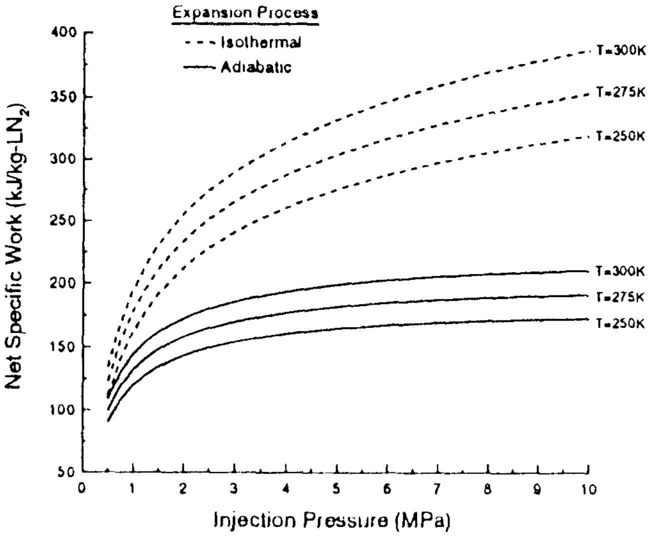

Fig. 3 **Specific work for adiabatic and isothermal expansion processes as function of peak cycle temperature and pressure. Exhaust pressure is 0.11 MPa and pump work is included.**

It is useful to compare the performance capabilities of the LN_2 propelled vehicle with two other EV concepts: General Motor's new electric car called the "Impact" and a Honda CRX that was converted to operate with an advanced electric propulsion system. The Impact is an optimized EV that utilizes advanced lead-acid batteries, special tires for low rolling resistance, and a streamlined body having a very low drag coefficient.[13] This two-seater car contains 400 kg of batteries to get a maximum range of 240 km between recharges and it has an effective operating cost of 5¢/km (8¢/mi).* The modified CRX uses a pack of 28 lead-acid batteries, which have a total mass of 500 kg and occupy a volume of 240 liters (63 gal). The maximum range for this EV at cruise is 180 km and its effective

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* Including cost of replacing batteries every 20,000 miles.

TABLE 1: Operating Parameters of the Ambient Air Powered LN_2 Automobile

Process:	Adiabatic	Isothermal
Pump Work:	6 kJ/kg-LN_2	6 kJ/kg-LN_2
Net Work Out:	194 kJ/kg-LN_2	314 kJ/kg-LN_2
Heat Input:	419 kJ/kg-LN_2	750 kJ/kg-LN_2
Energy Density:	54 W-h/kg-LN_2	87 W-h/kg-LN_2
LN_2 Flow Rate:†	1.5 kg/km	0.93 kg/km
Operating Cost:‡	3.9 ¢/km	2.4 ¢/km

† Based on 7.8 kW for highway cruise at 97 km/h.
‡ Based on 2.6¢ per kg-LN_2 production cost.

operating cost is 28¢/km (45¢/mi).** If the CRX were modified with an LN_2 propulsion system and the cryogen storage vessel occupied the same volume as the lead-acid batteries, then this vehicle would have the same range while saving 300 kg in energy storage mass.

Not only are the operating costs of the cryogen propulsion systems lower than those for the EV concepts discussed above, they are also competitive with the advanced battery systems currently being developed. As can be seen in Fig. 4, the LN_2 storage system compares favorably with the lead-acid battery on a per mass basis. Indeed, if near-isothermal performance can be achieved, the specific energy characteristics become particularly attractive. In addition, since the fully loaded LN_2 tank would comprise about 25% of the total vehicle mass (30% is typical mass ratio for lead-acid battery systems),[14] the cryomobile performance should increase as the cryogen is consumed.

IV. COMPONENT DESCRIPTIONS

Expander

The maximum work output of the LN_2 engine results from an isothermal expansion stroke. Achieving isothermal expansion will be a challenge, because the amount of heat addition required during the expansion process is nearly that required to superheat the pressurized LN_2 prior to injection. Thus, engines having expansion chambers with high surface-to-volume ratios are favored for this application. Rotary expanders such as the Wankel may also be well suited. A secondary fluid could be circulated through the engine block to help keep the cylinder walls as warm as possible. Multiple expansions and reheats can also be used although they require more complicated machinery.

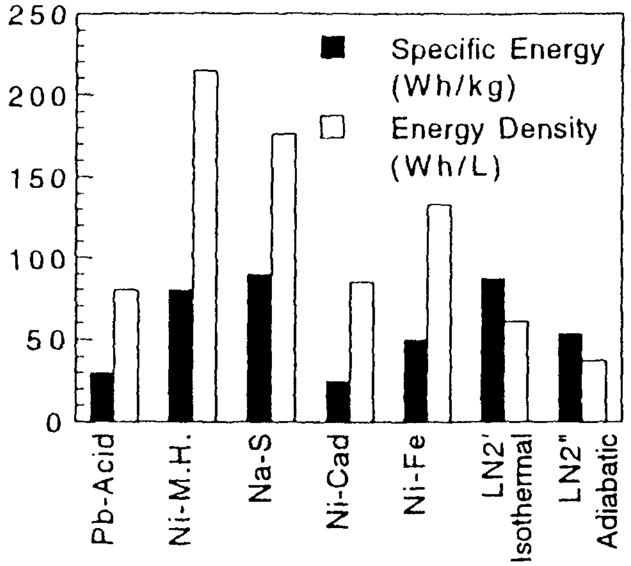

Fig. 4 **Energy densities of various electrochemical batteries compared with LN_2 thermal storage. LN_2 is expanded from 300 K and 4 MPa to 0.11 MPa.**

Vehicle power and torque demands would be satisfied by both throttling the mass flow of LN_2 and by controlling the cut-off point of N_2 injection, which is similar to how classical reciprocating steam engines are regulated. The maximum power output of the propulsion engine is limited by the maximum rate at which heat can be absorbed from the atmosphere. The required control system to accommodate the desired vehicle performance can be effectively implemented with either manual controls or an on-board computer. The transient responses of the LN_2 power plant and the corresponding operating procedures are topics to be investigated.

Heat Exchanger

The primary heat exchanger is a critical component of a LN_2 automobile. Since ambient vaporizers are widely utilized in the cryogenics and LNG industries, there exists a substantial technology base.[15] Unfortunately, portable cryogen vaporizers suitable for this new application are not readily available at this time. To insure cryomobile operation over a wide range of weather conditions, the vaporizer

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** Including cost of replacing batteries every 1,500 miles.

should be capable of heating the LN_2 at its maximum flow rate to near the ambient temperature on a cold winter day. Since reasonable performance for personal transportation vehicles can be obtained with a 30 kW motor, the heat exchanger will be sized accordingly. For an isothermal expansion engine having an injection pressure of 4 MPa, the heat absorbed from the atmosphere can, in principle, be converted to useful mechanical power with about 40% efficiency. Thus the heat exchanger system should be prudently designed to absorb at least 75 kW from the atmosphere when its temperature is only 0°C.

To estimate the mass and volume of the primary heat exchanger, it was modeled as an array of individually fed tube elements that pass the LN_2 at its peak flow rate without excessive pressure drop. Each element is a 10 m long section of aluminum tubing having an outside diameter of 10 mm and a wall thickness of 1 mm. They are wrapped back and forth to fit within a packaging volume having 0.5 m x 0.4 m x 0.04 m dimensions and are arrayed in the heat exchanger duct as shown in Fig. 5. Incoming air will pass through a debris deflector and particulate filter before encountering the elements. An electric fan will draw the air through the duct when the automobile is operating at low velocities or when above normal power outputs are required.

The tube exterior heat transfer coefficient is based on that for a cylinder in cross flow and the internal heat transfer is for fully developed turbulent flow.[16] The bulk temperature of the air is assumed to decrease across each tube row as determined from energy conservation and the pressure drop is determined for the whole tube bank. The heat transfer calculations also account for N_2 pressure drop and variations in its thermodynamic properties in the tube elements.[17] Some of the important phenomena not considered at this stage of analysis were the effects of transient LN_2 flow rates, start up, frost accumulation, tube fins, and axial thermal conduction.

To illustrate the operational characteristics of this ambient vaporizer, temperature profiles along a single element exposed to 300 K air moving at 18 km/h (11 mi/h) are shown in Fig. 6. The LN_2 and air mass flow rates are 8 gm/sec and 0.24 kg/sec, respectively. For this example the peak cycle temperature is 260 K and the air stream exits the heat exchanger at about 250 K. Since the in-tube heat transfer coefficient is very high, the tube wall remains close to the LN_2/N_2 temperature, even though the cryogen flow rate is much less than that of the outside air. Thus the formation of rime ice is highly probable for these conditions. The atmospheric moisture will be removed relatively quickly as the ambient air is

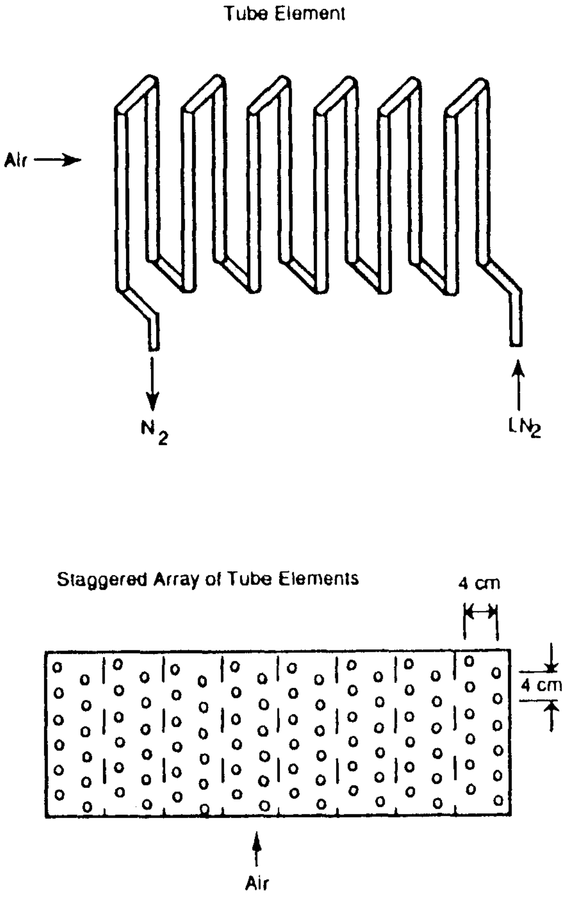


Fig. 5 Schematic of heat exchanger element used in analysis (top) and staggered tube array for determining air pressure drop (bottom).

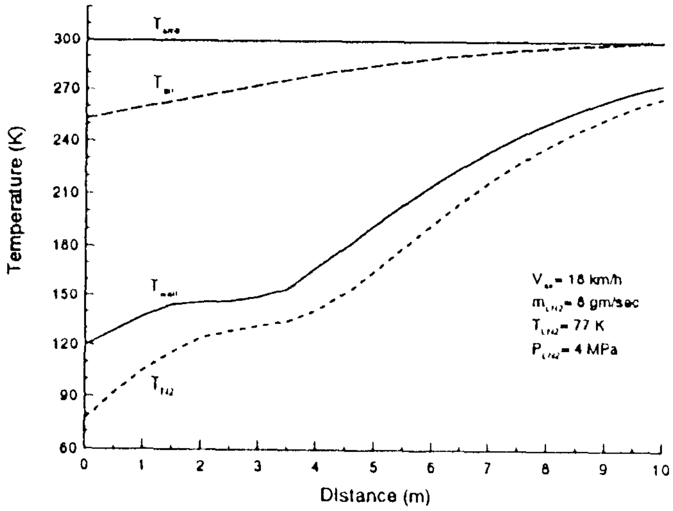
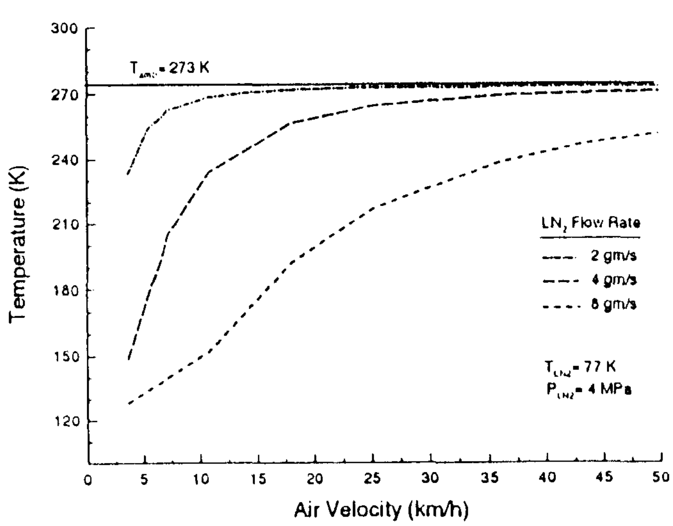


Fig. 6 Temperature profiles of a single heat exchanger element. Aluminum tubing: 8 mm I.D. x 10 mm O.D. x 10 m long.

chilled over the first few tube rows, leaving extremely dry air to warm up the coldest parts at the rear of the heat exchanger where the LN_2 enters. Surface coatings such as Teflon can be used to inhibit ice build up and active measures for vibrating the tube elements may also be applied. However, these approaches may not be necessary since high LN_2 flow rates are only needed during times of peak power demand and the heat exchanger elements are much longer than necessary to elevate the LN_2 temperature to near ambient at the lower flow rates required for cruise. Thus, the frosted tube rows may have ample opportunity to de-ice once the vehicle comes up to speed.



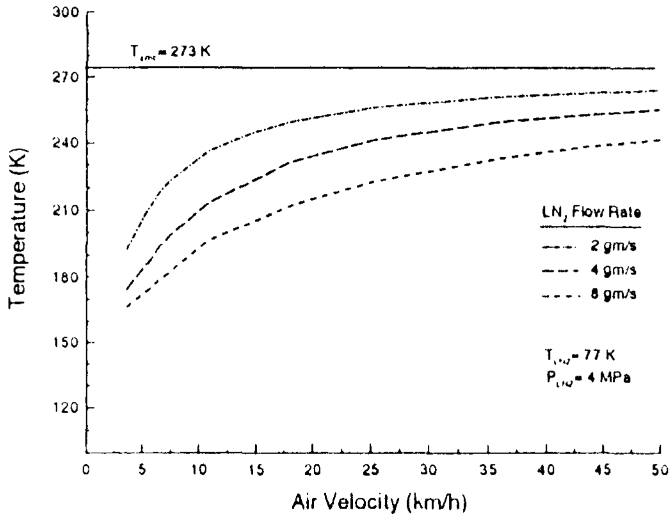


Fig. 7 N_2 temperature (top) at exit of tube element and air temperature (bottom) at exit of heat exchanger duct for different LN_2 flow rates as function of incoming air velocity.

The effects of the relative air velocity on the N_2 temperature at the exit of a single tube element and the air temperature at the exit to the heat exchanger are shown in the top and bottom plots, respectively, of Fig. 7 for several different LN_2 flow rates. Ambient temperature was 273 K and the N_2 injection pressure was 4 MPa for these calculations. It is apparent that the relative air velocity must be over 50 km/h (31 mi/h) to heat the 8 gm/sec flow to a reasonable operating temperature (240 K), whereas the 2 gm/sec flow will exit the heat exchanger element at ambient temperature if the vehicle speed is over 18 km/h. Correspondingly, the temperature drop in the air flow decreases significantly at higher velocities, indicating the potential for cruising at highway speeds on a cold day without incurring an extraordinary icing problem. The propulsion control system would insure that adequate amounts of air were drawn through the heat exchanger to meet power demands at any velocity and to minimize frost build up.

Even though inclement weather will certainly degrade the performance of the cryomobile, it will not preclude effective operation. If the propulsion system operating conditions were such that the LN_2 could only be heated to 250 K prior to injection, the flow rates of LN_2 for the isothermal and adiabatic cycles to generate 30 kW would be 115 gm/sec and 187 gm/sec, respectively. The previously described heat exchanger configuration can theoretically heat the higher LN_2 flow rate to 250 K with 25 radiator elements when the vehicle is traveling at 25 km/sec (16 mi/h) and the ambient air temperature is only 0°C (Fig. 7). The LN_2 viscous pressure drop would be about 0.05 MPa, which is easily compensated for with the cryogen pump. The electric fan would require approximately 1.5 kW to accelerate the air and overcome the 400 Pa pressure drop through the heat exchanger if the vehicle were standing still. Since each element is 0.76 kg, the total tubing mass would be 19 kg. If the same mass was added by the manifolds and duct then the net mass of the heat exchanger would be under 40 kg. When operating on a typical California day, it is expected that this over-designed cryogen vaporizer will readily heat the LN_2 up to ambient temperature without any appreciable icing.

Cryogen Storage Vessel

The primary design constraints for automobile cryogen storage vessels are: resistance to deceleration forces in the horizontal plane in the event of a traffic accident, low boil-off rate, minimum size and mass, and reasonable cost. Crash-worthy cryogen vessels are being developed for hydrogen-fueled vehicles that will prevent loss of insulating vacuum at closing speeds of over 100 km/h.[18] Moderately high vacuum (10^{-4} torr) with super insulation can provide boil-off rates as low as 1% per day in 200 liter (53 gal) containers. Using appropriate titanium or aluminum alloys for the inner and outer vessels, a structurally reinforced dewar could readily have a seven-day holding period. The cost of a mass produced, 200 liter automotive tank for liquid hydrogen containment has been estimated to be between \$200 and \$400 (in 1970 dollars). Thus the expense of a 400 liter LN_2 tank (or two 200 liter tanks) is expected to be reasonable.

V. TECHNICAL ISSUES

Range Extension and Power Boosting

Range extension and performance enhancement can be realized by heating the LN_2 to above ambient temperatures with the combustion of a relatively low pollution fuel such as ethanol or natural gas. The augmentation of power output is most apparent for the adiabatic expansion engine, as can be seen in Fig. 8. By increasing the gaseous N_2 temperature to 500 K, the specific work at 4 MPa for the adiabatic engine is increased by 60% to make it nearly the same as the work from an isothermal expansion engine operating at 300 K. To evaluate the performance enhancement potential for an isothermal engine, the high temperature N_2 is assumed to polytropically expand to the end state reached when the engine is operating isothermally at ambient temperature. In this particular propulsive cycle an extra superheat of 200°C results in only a 30% increase in specific power. Thus the advantage of operating above ambient temperature depends, in part, on how isothermal the expansion process can be made to be.

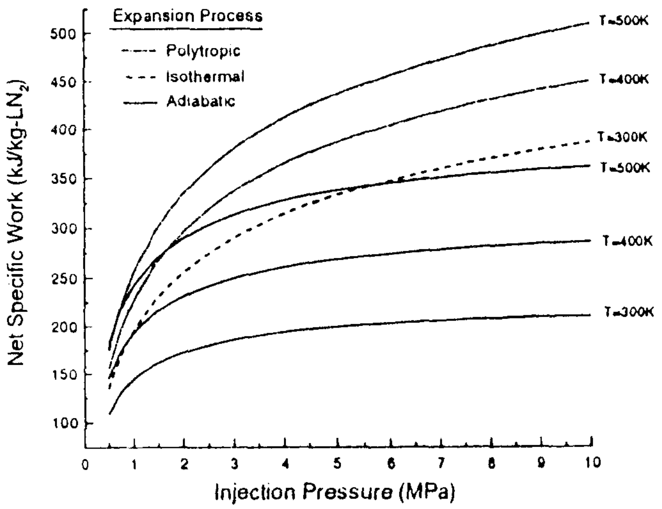


Fig. 8 **Specific work for adiabatic and polytropic expansion to 0.11 MPa from temperatures above ambient. N_2 temperature is 300 K after polytropic expansion.**

There is also the intriguing possibility of storing energy for boosting power or extending range by applying a medium that undergoes a phase change to the final superheater segment of the heat exchanger system. Ideally the phase change material would be slowly "recharged" as it absorbs heat from the atmosphere while the vehicle is parked and during cruise when peaking power is not required. Fast recharging with electric heaters may also be considered. We recognize that this added complexity must compete in mass and compactness with the alternative of just carrying more LN_2.

While extremely cold weather would degrade performance of a LN_2 propulsion system, this would not diminish the cryomobile's advantage over most battery EVs since their performance is severely compromised in cold weather. Below freezing, air temperatures are extremely warm to the cryogen "fuel" and the moisture content of the atmosphere is significantly diminished. Thus it is anticipated that sufficient enthalpy can still be drawn from the air to provide ample power without incurring a detrimental icing penalty. If necessary, an auxiliary combustor can be added that would allow continuous use of the vehicle on the very coldest days.

Liquid Nitrogen Infrastructure Considerations

There are many advantages that might be anticipated pending successful development of a transportation system based on liquid nitrogen. Large scale demand for LN_2 would result in some inevitable cleansing of the atmosphere due to the air liquefaction process. Atmospheric contaminants which have proved troublesome in highly congested environments, such as sulfur and nitrogen compounds, chlorinated fluorocarbons, and other greenhouse gases, are necessarily removed during the air liquefaction process and can be readily disposed of in an environmentally friendly manner.[19,20] Further expansion of a liquid nitrogen infrastructure would also have the effect of promoting new technologies that can enhance the efficiency of operations natural to a big city, involving everything from air-conditioning and food processing to industrial processes which currently use various components of air in large quantities.

The disadvantages of initiating a LN_2 based transportation system have to be assessed carefully.

The round trip energy efficiency of a practical LN_2 thermal storage system is less than that of batteries, approximately 17% for the isothermal example discussed here as compared to 60% for a typical EV.[14] However, the per-kilometer operating costs are much lower for the LN_2 car because there are no major component replacements (such as battery packs) that are required periodically. Many of the electrochemical battery systems under development require special recharging hardware that will require a substantial infrastructure investment before they can be widely implemented. Likewise, a cryogenic nitrogen distribution system will have to be developed. The LN_2 could either be produced on the dispensing site or trucked in as is done with conventional fuels. Due to its safety, above ground storage in vertical insulated tanks is convenient. Refilling operations for the cryomobiles will only take minutes and can be very safe. A prototype cryogen refueling station for passenger cars has been constructed to demonstrate the feasibility of transferring liquid hydrogen.[21] It is anticipated that adapting this technology to LN_2 distribution will not pose any significant technical challenges. Delivering road emergency quantities of LN_2 may also be much easier and more cost effective than carrying the electric charging equipment and spare batteries required for typical EVs.

VI. SUMMARY

Applying LN_2 as a portable thermal storage medium to propel both commuter and fleet vehicles appears to be an attractive means to meeting the ZEV regulations soon to be implemented. Pressurizing the working fluid while it is at cryogenic temperatures, heating it up with ambient air, and expanding it in reciprocating engines is a straightforward approach for powering pollution free vehicles. Ambient heat exchangers that will not suffer extreme icing will have to be developed to enable wide utility of this propulsion system. Since the expansion engine operates at subambient temperatures, the potential for attaining quasi-isothermal operation appears promising. The specific energy densities of LN_2 are 54 and 87 $W\text{-}h/kg\text{-}LN_2$ for the adiabatic and isothermal expansion processes, respectively, and the corresponding amounts of cryogen to provide a 300 km driving range would be 450 kg and 280 kg. Many details of the application of LN_2 thermal storage to ground transportation remain to be investigated; however, to date no fundamental technological hurdles have yet been discovered that might stand in the way of fully realizing the potential offered by this revolutionary propulsion concept.

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Frost-Free Cryogenic Heat Exchangers for Automotive Propulsion

J. Williams, C. Knowlen, A.T. Mattick and A. Hertzberg
Aerospace and Energetics Research Program
University of Washington
Seattle, WA

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FROST-FREE CRYOGENIC HEAT EXCHANGERS FOR AUTOMOTIVE PROPULSION

J. Williams,* C. Knowlen,† A.T. Mattick,†† and A. Hertzberg‡
Aerospace and Energetics Research Program
University of Washington, Box 352250
Seattle, WA 98195-2250

The University of Washington is studying a concept in the field of zero-emission vehicles: the cryogenic automobile. The cryogenic automobile uses nitrogen, stored in liquid state, as its working fluid. The liquid nitrogen is pressurized, then vaporized in an economizer through which the exhaust from the expander runs in counterflow. The resulting gaseous nitrogen is superheated in an ambient-air heat exchanger. This produces a high pressure, near-ambient temperature gas which is injected into a quasi-isothermal expander to produce the system's motive work. The spent, low pressure nitrogen is exhausted back to the atmosphere. One technical challenge which must be overcome is designing a heat exchange system that is structurally robust, works well under a variety of operating conditions and is not hampered by the buildup of frost. The formation of frost on sub-ambient heat exchangers increases conductive resistance to heat transfer, occludes air-side flow passages, and contributes to structural loading. A concept is described by which vaporization and superheating of the nitrogen is achieved in a frost-free manner. This is accomplished without the use of mechanical, electrical or thermal frost removal devices. This paper presents the design and fabrication of such a heat exchange system.

Nomenclature

A = area (m^2)
c_p = specific heat (J/kg K)
F = heat flux (W/m^2)
h = specific enthalpy (J/kg)
Δh = change in specific enthalpy (J/kg)
k = thermal conductivity (W/m K)
L = element length (m)
$\dot{m}$ = mass flow (kg/s)
n_t = number of tube passes
N_{tu} = number of transfer units
P = perimeter (m)
Q = heat transfer rate (W)
r = radius (m)
T = temperature (K)
$\bar{U}$ = overall heat transfer coefficient (W/m^2 K)
ε = heat transfer effectiveness
η = non-dimensional span (x/L)
τ = non-dimensional temperature (T/T_{am})

Subscripts

a = ambient
c = condensation
$econ$ = economizer
LN_2 = liquid nitrogen
N_2 = gaseous nitrogen
o = initial
t = tube side
s = shell side
1 = inlet
2 = outlet

I. Introduction

In September of 1990, in an effort to improve local air quality, the California Air Resources Board enacted the Low Emission Vehicle (LEV) program.[1] The LEV program established several categories of emission standards for cars and light trucks. The most stringent of these categories was for the zero-emission vehicle (ZEV). The LEV program requires that, by 2003, each of the seven largest automobile manufacturers (Chrysler, Ford, General Motors, Honda, Mazda, Nissan and Toyota) produce and offer for sale ZEVs at a rate equal to 10% of the automobile sales each company has in the state, or about 110,000 cars per year.[2] Similar mandates have also been adopted by New York and Massachusetts.

The impetus for the LEV legislation is the desire to reduce air pollution. In urban areas of southern California, vehicles account for over 50% of the air pollution emitted.[3] In 1995, the South Coast Basin (which includes Los Angeles, Orange, and parts of San Bernardino and Riverside counties) experienced 98 days in which the EPA health standard for ground-

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* Graduate Research Assistant, Student Member AIAA
† Research Scientist, Senior Member AIAA
†† Professor, Member AIAA
‡ Professor Emeritus, Fellow AIAA

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level ozone was exceeded. Ground-level ozone can cause aching lungs, wheezing, coughing and headaches. Serious health problems can also arise for those people with asthma, emphysema and chronic bronchitis. Children appear to be at particular risk. A 1984 study conducted at USC showed that children raised in the South Coast Basin suffered a 10% to 15% decrease in lung function.[4]

The deleterious effects of gasoline and diesel powered vehicles are not limited to air quality in southern California. In half of the world's cities, tailpipe emissions are the single largest source of air pollution. Worldwide, automobiles account for half of the oil consumed and a fifth of the greenhouse gases emitted. This situation is not expected to improve in the near future, as the number of cars and light trucks in the world – over 500 million – is expected to double in the next thirty years. Most of this growth will occur in developing countries which have little or no emission controls.[5]

Currently, the battery powered electric vehicle is the only commercially available technology that can meet ZEV standards. However, electric vehicles have not sold well. This is primarily due to their limited range, although anemic performance, slow recharge and high initial costs are also contributing factors. All of these issues can be traced directly to the limitations of electrochemical energy storage, particularly lead-acid batteries. Lead-acid remains the dominant technology in the electric vehicle market, but only exhibit energy densities in the range of 30-40 W-hr/kg. This compares with about 3,000 W-hr/kg for gasoline combusted in an engine running at 28% thermal efficiency.[6] Lead acid batteries can take hours to recharge and must be replaced every 2–3 years. This raises the specter of increased heavy metal pollution, were a lead-acid powered electric fleet ever to come to pass.[7]

Even advanced battery systems, such as nickel-metal hydride, zinc-air, and lithium-ion suffer from slow recharge and high initial cost. Nickel-metal hydride batteries, often touted as the heir-apparent of lead-acid, still contain a heavy metal and must realize dramatic reductions in cost in order to be truly competitive. Lithium-ion batteries, considered by many to be the third-generation solution, must also contend with cost and demonstrate their safety to a wary public.

Another energy storage medium will be required to make ZEVs the *non-mandated* automobile choice of the car-buying public. If certain technical challenges can be overcome, that energy storage medium may well be liquid nitrogen. Since 1993, the University of Washington has been researching the technical challenges involved in building and operating a vehicle powered by liquid nitrogen.[8] Issues pertaining to frost-free heat exchanger performance, cryogenic equipment, cycle analysis, drive train selection and vehicle configuration are being investigated.

This paper describes the fundamental concepts of cryogenic automotive propulsion. This includes the thermodynamic theory, the individual system components, the production of liquid nitrogen and infrastructure considerations. The analysis, design, and fabrication of a two-stage liquid nitrogen heat exchange system is discussed in detail. The experimental facilities, consisting of a low-speed wind tunnel and a prototype vehicle, are also described. Finally, conclusions and recommendations for future research are presented.[9]

II. Cryogenic Automotive Propulsion

The cryogenic automobile is a zero-emission vehicle. It operates on the thermodynamic potential between the ambient atmosphere and a reservoir of liquid nitrogen. One way to utilize that potential is through an open Rankine cycle. The liquid nitrogen is drawn from a tank, pumped up to the system pressure, then vaporized and superheated in a two-stage heat exchange system. The resulting high pressure, near-ambient temperature gas is injected into a quasi-isothermal expander which produces the system's motive work.[10] The spent, low pressure gas is exhausted back to the atmosphere.[8] Because a zero emission vehicle is required to produce no smog-forming tailpipe or evaporative pollutants,[11] and because nitrogen gas is its only emission, the cryogenic automobile meets California's ZEV guidelines.

Although the concept of a nitrogen powered automobile has been studied in the past,[12,13] there are two key technologies that have yet to be demonstrated: the quasi-isothermal expander and a frost-free liquid nitrogen heat exchange system. It is this heat exchange system that is the primary subject of this paper

Overview

There are many thermodynamic cycles available for utilizing the thermal potential of liquid nitrogen. These range from the Brayton cycle, to using two- and even three-fluid topping cycles, to employing a hydrocarbon-fueled boiler for superheating beyond atmospheric temperatures. The easiest to implement, however, and the one chosen for this study, is shown in Fig. 1. This system uses an open Rankine cycle. It begins with a tank of liquid nitrogen stored at 77 K and 1 bar. The nitrogen is pumped, as a liquid, to the system's working pressure. This high pressure liquid flows into the economizer. The economizer is a shell-and-tube heat exchanger where the shell-side fluid is the exhaust from the expander. While this step is not necessary from an energy point of view, it does have the advantage of providing a frost-free pre-heat to the incoming liquid.

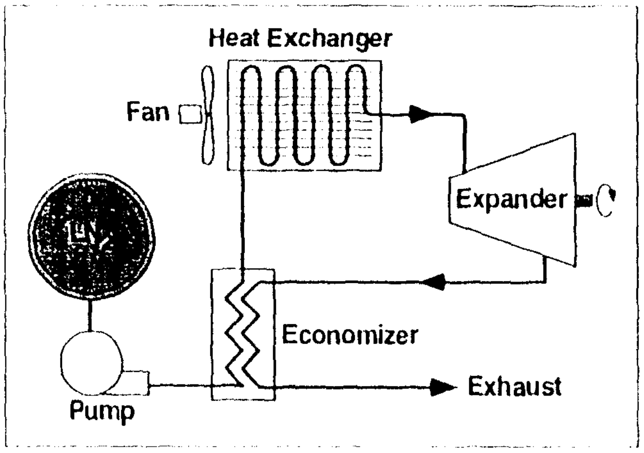

Fig. 1: Liquid nitrogen propulsion system.

Once through the economizer, the vaporized nitrogen enters the heat exchanger, which has a multi-element, tube-in-crossflow configuration. The exterior fluid is the ambient atmosphere, which is drawn through the core of the heat exchanger either by the motion of the vehicle or by a fan, depending on the operating regime. This heat exchanger must be able to operate across most of the spectrum of environmental and operating conditions without suffering the adverse effects of frost buildup.[14]

Upon leaving the heat exchanger, the working fluid is a high pressure, near-ambient temperature gas. It is injected into the expander which provides all of the motive work for the system. This can be either a positive displacement or turbine engine. Following expansion, the low-pressure exhaust is warm enough to be used in an economizer, where it preheats the incoming liquid, before finally being vented to the atmosphere.

The temperature-entropy diagram for the open Rankine cycle, operated at critical pressure, is shown in Fig. 2. Labels 1-2 indicate the pumping process. Because pressurization is occurring in the liquid phase of the fluid, the work required is small in comparison with the available work. Process 2-3 is the pass through the economizer and heat exchanger. Process 3-4 and 3-4' are the isothermal and adiabatic modes of expansion, respectively. If the shaded area represents the available specific work in the cycle, then these two processes provide the upper and lower limits to the expander's performance. Process 4-1 (or 4'-1) is the liquefaction stage. This occurs remotely at an air processing plant.

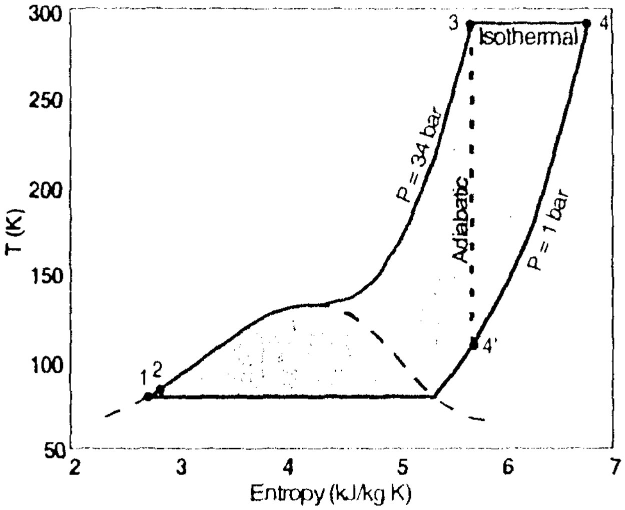

Fig. 2: Temperature-entropy diagram for the open Rankine cycle.

The thermodynamic and economic performance of the adiabatic and isothermal modes of the open Rankine cycle are shown in Table 1. These figures are based on the specifications of a modified Honda CRX for which performance data were available.[15] The cost of 2.6¢ per kg-LN_2 was derived assuming only the energy cost of production.

Table 1: Performance of the open Rankine cycle.[8]

Process	Adiabatic	Isothermal
Pump Work:	6 kJ/kg-LN_2	6 kJ/kg-LN_2
Net Work Out:	194 kJ/kg-LN_2	314 kJ/kg-LN_2
Heat Input:	419 kJ/kg-LN_2	750 kJ/kg-LN_2
Energy Density:	54 W-h/kg-LN_2	87 W-h/kg-LN_2
LN_2 Flow Rate:†	1.5 kg/km	0.93 kg/km
Operating Cost:‡	3.9 ¢/km	2.4 ¢/km

† Based on 7.8 kW for highway cruise at 97 km/h.
‡ Based on 2.6¢ per kg-LN_2 production cost.

Figure 3 shows how a liquid nitrogen based propulsion cycle fares against the various electrochemical storage media mentioned earlier. Specific energy is a useful figure of merit because it correlates closely with range. Even the next generation, nickel-metal hydride battery, only matches the performance of the isothermal open Rankine cycle. And the open Rankine is not the highest performing cycle available. By adding a methane topping cycle, upwards of 160 W-hr/kg can be achieved.

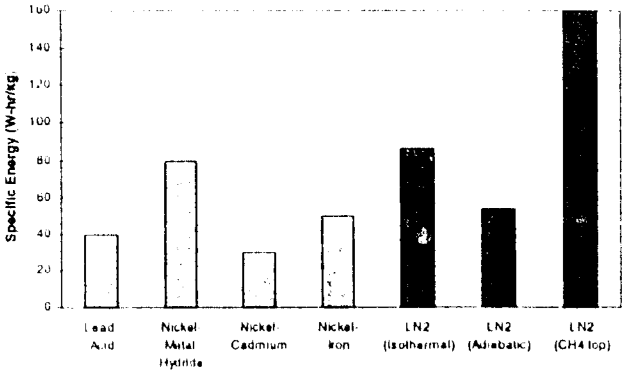

Fig. 3: Specific energy for various energy storage media.

Comparisons of specific energy provide only a partial view of how a vehicle will be accepted by the car-buying public. According to a 1997 J.D. Power and Associates survey titled "Electric Vehicles: The Consumer Perspective",[16] American drivers cited numerous factors they felt were important in determining which vehicle to purchase. While these included everything from cost to styling, conspicuously absent was any reference to environmental impact. Tim Gohmann, director of custom research at J.D. Power and Associates, put it more succinctly: "The true success of the EV will be based on its performance as a vehicle, not as an environmental solution."[17] It is reasonable to assume the same argument applies to cryogenic automobiles.

Liquid Nitrogen Production and Infrastructure

Having a liquid nitrogen infrastructure in place is crucial for the cryogenic automobile to become a mass-market vehicle. Yet that infrastructure will have little reason to exist until such a market is established. This chicken-and-egg argument is central to the acceptance of a nitrogen powered automobile as a viable transportation option. Liquid nitrogen is currently produced on large scales for industrial, institutional, military, aerospace and medical applications; however, it is not distributed widely, as is electricity, gasoline and diesel fuel.

Several options are possible for the distribution of liquid nitrogen. These range from the commonplace, such as trucking, to the ambitious, such as piping it through insulated jackets wrapped around power transmission lines. A more obvious solution is to not transport it at all. Installing a small-scale liquefaction facility at the site of a filling station has many advantages, not the least of which being that it can tap into the local power grid for its electricity requirements. The economics of this concept require further study.

Large scale liquid nitrogen production has its advantages as well. Figure 4 illustrates a concept in which the nitrogen liquefaction process is driven by a natural gas-fired power plant. Rather than operating on the ambient air, the liquefaction plant uses the exhaust from the gas turbine. In the process of obtaining the nitrogen, oxygen (a marketable commodity), is also liquefied. Moreover, the carbon dioxide from the power cycle combustion process can be condensed from the exhaust stream. A modern combined-cycle, gas turbine power plant can produce enough energy to liquefy up to 70% of its own exhaust – most of which is nitrogen – while sublimating virtually all of the CO_2.[8] This can then be disposed of in several ways less harmful to the environment than simply venting it to the atmosphere.[18,19,20] This process raises the possibility of creating a liquid nitrogen-based transportation infrastructure that produces no atmospheric CO_2 emissions, and is the subject of further study.

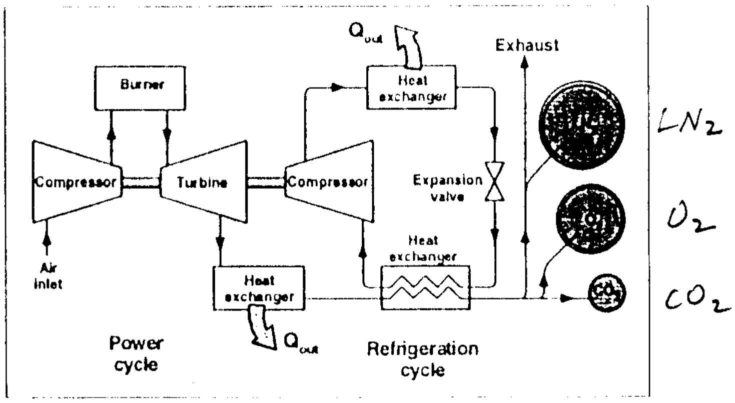

Fig. 4: Proposed nitrogen liquefaction process.

III. Experimental Facilities

Wind Tunnel

The University of Washington's Cryogenic Energy Storage Laboratory is equipped with a low speed wind tunnel ($0 \leq V_\infty \leq 5$ m/s), which has been fitted with a 20 cm × 50 cm Plexiglas* test section which can accommodate a variety of heat exchangers in cross-, parallel- or counter-flow arrangements. As shown in Fig. 5, the liquid nitrogen is fed into the system from a 160 liter stainless steel dewar. The LN_2 is kept under a constant 34 bar pressure blanket of nitrogen gas from regulated gas bottles.[14]

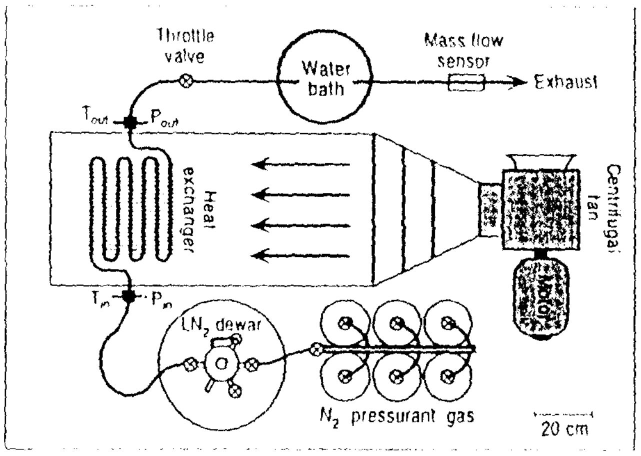

Fig. 5: Low-speed wind tunnel facility.

The liquid nitrogen is plumbed through the test section, where the flow temperature and pressure are measured at both the inlet and the outlet. The exiting gas is then throttled down to atmospheric pressure and piped through a liquid water heat bath. This raises the gas temperature to ambient just prior to entering the mass flow meter, after which it is exhausted through the laboratory's normal ventilation system.

Data are collected and stored on a computer at a sample rate of 30 data points per minute. Also recorded are the tunnel speed, ambient temperature and ambient humidity. Frost depth and accumulated mass can also be measured.

Test Vehicle

A test vehicle has been purchased to serve both as a proof-of-concept and as a rolling test-bed for further system refinements. The vehicle itself is a 1984 Grumman-Olson Kubvan, pictured in Fig. 6. To emphasize the near-term potential of this project, it has been christened the *LN2000*. This model vehicle was originally built for the US Postal Service and could be purchased with either a diesel or electric drivetrain. This particular vehicle was originally electric, operating on a pack of 14 lead acid batteries which weighed over 450 kg. The running gear for this vehicle is from a right-hand drive 1984 Volkswagen Rabbit. The construction is welded frame with riveted body panels and is made entirely of aluminum.

Fig. 6: 1984 Grumman-Olson Kubvan.

The Kubvan was selected as a test-bed for several reasons. The volume available is well suited to the placement of the necessary equipment. The simplicity of its construction - flat sheet-metal body panels, aluminum frame, open interior - is conducive to making modifications. Also, because it was originally designed to be electric, it operates with very few "hotel" functions such as air conditioning and rear window defrosting.

The Kubvan performance using the prototype LN_2 propulsion system has been calculated and is given in Table 2. This employs an EPA approved Federal Urban Driving Schedule. Velocity dependent rolling resistance and aerodynamic drag were calculated,[21] and correlated reasonably well with coast-down tests made with the vehicle.[22]

Table 2: Performance of the open Rankine cycle.[8]

	City Cycle	Highway Cycle	Straight and Level*
Avg. Mass Flow (g/s)	81	204	256
Max. Mass Flow (g/s)	297	298	256
Consumption (kg/km)	9.5	10.0	10.8
Avg. Power (kW)	4.0	10.7	13.0
Max. Power (kW)	13.0	13.0	13.0

* At a top speed of 85 km/h (53 mph).

The equipment layout is shown in Fig. 7. Most of the cryogenic plumbing is stainless steel, but the components attached to the economizer are aluminum. Low pressure plumbing utilizes large diameter rubber hose where possible. Each of the items pictured is described below.

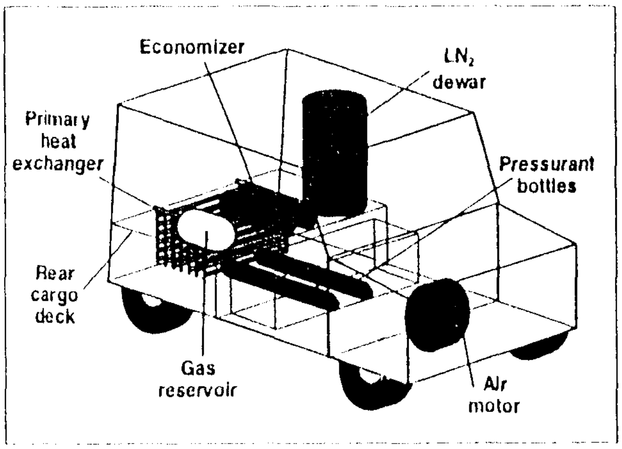

Fig. 7: Schematic of vehicle equipment layout.

Liquid Nitrogen Storage Tank: The dewar chosen for this application can hold 80 liters of liquid nitrogen at 24 bar with a daily boil-off rate of ~3%. The primary protection against over-pressure is a relief valve connected to the internal vessel. This valve also serves as the bleed for the boiloff gases, which are vented to the outside by a rubber hose. There are several other safety devices, providing multi-tiered protection against catastrophic rupture. The dewar is held in place at five attachment points: one on the roof, the other four on the floor.

Pressurization System: The pressurization system consists of two high-pressure nitrogen bottles stored under the rear deck of the Kubvan. The blowdown system has the advantage of mechanical simplicity at the cost of increased weight and volume. Each of the nitrogen bottles has a mass of 40 kg. The volume of gas required was calculated such that the pressurant tanks and the dewar get to within 3 bar of equilibrium just as the last of the liquid nitrogen is drained out. The pressurant tanks are initially filled to a starting pressure of about 133 bar. This is regulated down to the system pressure of 24 bar before being injected into the dewar. The hardware required for filling both the pressurant bottles and the dewar is attached to the vehicle.

Economizer: The economizer, which will be described in detail in Chapter IV, is actually a pair of shell-and-tube heat exchangers, as shown in Fig. 8. These heat exchangers operate in parallel, with the shell-side fluid being the exhaust from the expander. When operating at maximum mass flow, ~300 g/s, the economizer is designed to bring the liquid nitrogen to a quality of about 75%. This represents approximately one quarter of the total enthalpy change the nitrogen will experience before being injected into the expander. At lower mass flows, the vaporization will be complete.

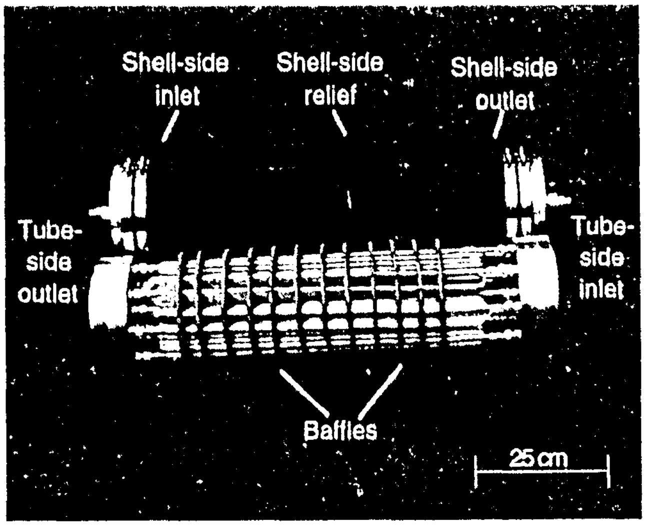

Fig. 8: Economizer units with and without shell.

Ambient-Air Heat Exchanger: The ambient-air heat exchanger, as will be detailed in Chapter V, is made up of 45 finned-tube elements. These elements are manifolded together, as shown in Fig. 9, to make a staggered array of tubes in crossflow with the incoming air. Either the motion of the vehicle, or the two ducted fans located at the back of the van, draw the air through the heat exchanger. The air inlet consists of a sheet-metal scoop slung underneath the vehicle.

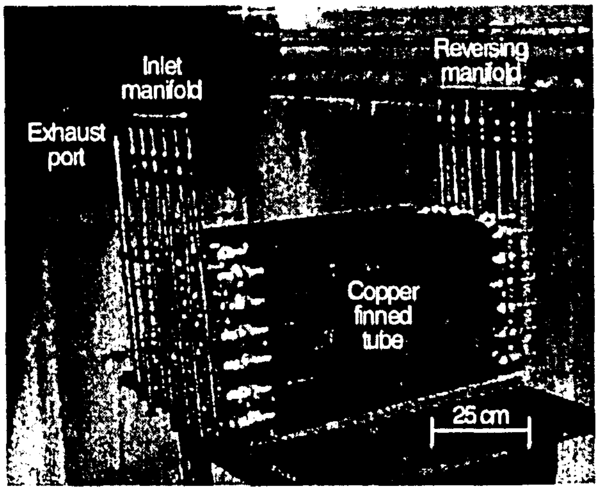

Fig. 9: Ambient-air heat exchanger assembly.

Expander: The expander chosen for the prototype vehicle is a 11.1 kW, radial piston air motor made by Cooper Power Tools. This motor, pictured in Fig. 10, has a cast-iron block with five 7.5 cm cylinders. Each cylinder holds a steel piston attached to the single-throw crank shaft by a connecting rod. Lubrication is maintained by splash and by an oiler located near the gas inlet. The motor is attached to the front-wheel drive, 5-speed manual transmission by a custom-made aluminum gear box. The output shaft of the motor drives a 15.24 cm diametral pitch (DP) spur gear. The input shaft to the clutch assembly has a 7.62 cm. DP spur gear, giving a 1:2 speed ratio through the gear box. The running gear is from a 1984 Volkswagen Rabbit and is right-hand drive.

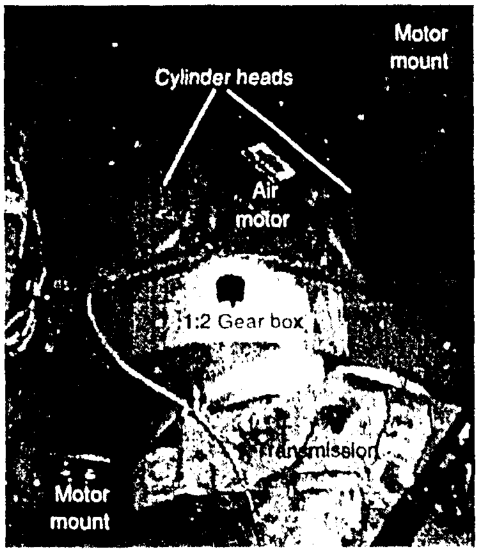

Fig. 10: Motor and transmission assembly.

IV. Development of the Economizer

As of this writing, the economizer has been built, pressure tested and installed in the prototype vehicle. Its performance as a heat exchanger has not yet been tested. Described herein is the theoretical modeling, design and fabrication of the economizer units. Future plans include rigorous road testing to verify the model and to identify possible modifications for a second generation economizer.

Design

The economizer is required to vaporize and, if possible, superheat liquid nitrogen at mass flow rates of up to 300 g/s. It must not produce excessive pressure drop on either the tube- or shell-side fluids. The economizer must also be safe in both normal operations and during an accident.

For the design of the economizer, a variant of the shell-and-tube heat exchanger has been employed. The tube-side fluid is the incoming liquid nitrogen and the shell-side fluid is the exhaust from the expander. To examine the theoretical limitations on the performance of the economizer, conservation of energy can be applied as shown in Eq. 1.

$$\dot{m}_{LN_2}\,(h_2 - h_1)_t = \dot{m}_{N_2}\,(h_2 - h_1)_s \qquad (1)$$

For steady state operation, $\dot{m}_{LN_2} = \dot{m}_{N_2}$, so the tube-side outlet conditions can be specified by

$$h_{2,t} = h_{1,t} + \left(h_2 - h_1\right)_s \qquad (2)$$

To calculate the maximum theoretical performance, assume that the outlet temperature of the shell-side gas is equal to the saturation temperature of the tube-side fluid (~120 K). If the exhaust pressure and temperature of the nitrogen gas is 3 bar, and 270 K respectively, then the maximum enthalpy at the tube-side exit of the economizer, h_{max}, is 140 kJ/kg, corresponding to a temperature of 155 K. This equates to over 47 kW of steady state heat transfer at the maximum mass flow of 300 g/s, and is 35 K above the saturation temperature of the tube-side fluid. This calculation provides the figure of merit for the performance of the economizer, the heat transfer effectiveness:

$$\varepsilon_{econ} = \left(\frac{h_2 - h_1}{h_{max} - h_1}\right)_t = \frac{\Delta h}{\Delta h_{max}} \qquad (3)$$

where h_1 is -18 kJ/kg and Δh_{max} is 158 kJ/kg. If more than 50% of maximum performance can be attained (i.e.: $\varepsilon_{econ} > 0.5$), then this should allow the ambient-air heat exchanger to be designed for single phase flow.

Constraints: Due to limited space in the vehicle, the economizer must necessarily be of compact design. As shown in Fig. 7, the two economizer shells are situated in the center of the cargo deck, toward the front of the vehicle. The installation of a second dewar is planned for behind the driver's seat, so the diameters of the two shells are constrained to be such that they can accommodate this addition. Even though the vehicle is of rugged construction and has good weight bearing capability, low mass components are installed where possible.

Configuration: A shell-and-tube heat exchanger is an obvious choice for compact, yet highly effective duty. The configuration chosen for the economizer is shown in Fig. 8. Though welding is the joining process of choice for tube-to-manifold connections in industrial scale heat exchangers,[23] the available aluminum tubing had a wall thickness of 0.7 mm. This is a difficult size to weld, particularly when using aluminum and when joining to a considerably more massive manifold. This led to the use of compression fittings. Because each tube end has to be accessible to a wrench, the fittings are widely spaced on the manifold, as shown in Fig. 11.

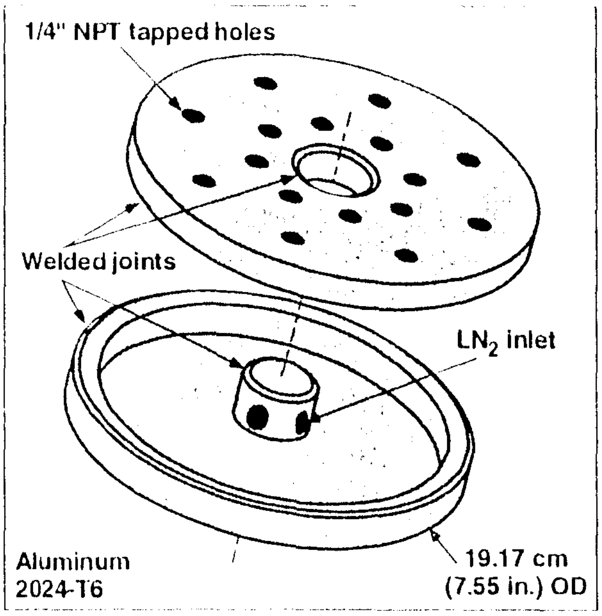

Fig. 11: Manifold showing hole pattern.

Due to this geometric constraint, the tube density for straight tubing is lower than optimum. To increase the length of tube per fitting, a multiple double-back configuration, shown in Fig. 12, was adopted. Each individual tube doubles back on itself four times between the inlet and the outlet, to give five 0.5 m lengths. Each tube has three passes that are in counterflow with the shell-side fluid and two that are in parallel. To achieve an even greater density, each tube was intertwined with its neighbor before installation. There are two economizer units, with 16 tubes in each, totaling 90 m of 6.35 mm diameter tubing. This gives an overall heat transfer area of 1.8 m^2.

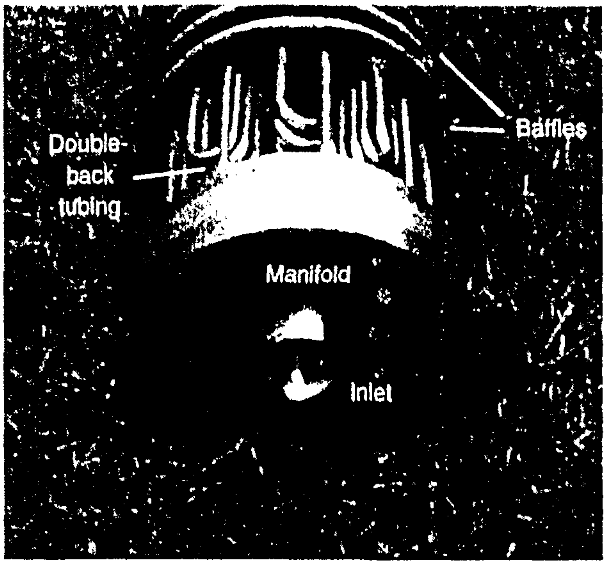

Fig. 12: Economizer unit showing double-back tube configuration.

Within the shells, the tubes are supported by twelve 3.2 mm rubber disks, or baffles, also shown in Fig. 8 and Fig. 12. These baffles form a seal with both the shell wall and the tubes themselves. Because they have flow passages cut on alternating sides, the baffles also serve to direct the shell-side gas over the tubing in a cross-flow pattern.

Materials and Fabrication: The manifolds are constructed of aluminum 2024-T6 and the tubing is aluminum 6061-T6. Aluminum, besides being readily available, has the advantages of low density, high conductivity, high yield strength at cryogenic temperatures, and excellent machinability. All fittings are either NPT or compression, and are made of aluminum, for comparable thermal expansion. Each shell is constructed of 25.4 mm thick, high-density polyethylene. The inlet, outlet and relief nipples are fused to the main body of the shell.

Structural Integrity: Each economizer is secured to the deck of the vehicle with two nylon straps wrapped around each shell. The shells themselves are pressure rated by the manufacturer to 8.5 bar. The shells are also equipped with one relief valve each, set at a gauge pressure of 4 bar. The aluminum tubing is pressure rated by the manufacturer to near 500 bar.

Each of the four manifolds is designed to withstand an internal pressure of 27 bar, at 77 K. Utilizing the plate bending correlations of Timoshenko,[24] and the yield strength for low-temperature aluminum given by Scott,[25] a factor of safety of 6.7 was obtained. Modifying this result with the ligament efficiency factor given by Harvey for perforated plate stiffness,[26] the final safety factor for the manifolds is 3.4.

When the economizer is operating, the shell will be pressurized to 3 bar. The manifolds will experience a differential pressure of 2 bar acting outward. Because of this, the economizers will see tensile forces in the axial direction of up to 6670 N during normal operations, and 21,600 N in the event of a catastrophic blow-by of the valving in the expander. To counter this, each economizer manifold is retained in place by aluminum brackets bolted together with two 1.9 cm threaded rods.

Modeling

A model was developed for the performance of the economizer, to serve as a design tool. This enabled informed decision-making about such geometric parameters as tube length, number of passes, number of baffles, etc. Pressure drop was not considered as one of the design variables because a blow-down system is being used. It takes no energy and very little additional mass to offset pressure drop in the tube-side of the economizer. For this reason, pressure drop correlations were not included in the model, however, the capability to include them at a later date is built in.

The unique flow arrangement of this heat exchanger does not lend itself to analysis by the ε-N_{tu} method.[27] Therefore, thermodynamic and heat transfer performance calculated numerically, using a finite differencing scheme. To model the economizer, both the shell-side and tube-side variables were discretized, according to the scheme shown in Fig. 13.

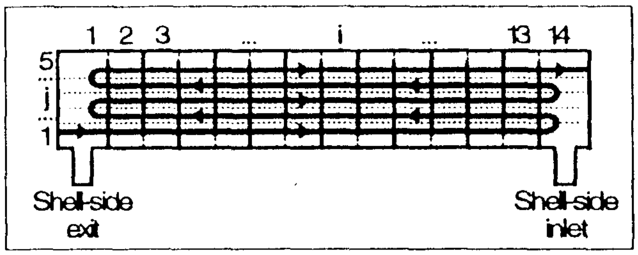

Fig. 13: Element numbering for one economizer tube.

The following assumptions were made in the modeling of the economizer:

- Thermodynamic equilibrium between fluid phases.
- Even flow distribution. There are 32 tubes and two shells, but only one of each is modeled.
- Steady-state operation.
- Fully developed flow.
- Adiabatic shell wall. This is also conservative because heat transfer from the air to the shell-side fluid will increase performance.
- Zero radiation.
- Zero axial conduction.

Across each element of the economizer, an energy balance is performed. That is,

$$c_p\left(T_i - T_{i+1}\right)_s = n_t\left[\sum_{j=1,3,5}\left(h_{i+1,j} - h_{i,j}\right)_t + \sum_{j=2,4}\left(h_{i,j} - h_{i+1,j}\right)_t\right] \quad (4)$$

Note that for this element numbering system, the flow is left-to-right for passes 1, 3 and 5 and right-to-left for passes 2 and 4. For two-phase flow, the heat equation can be written

$$\overline{U}A\left(T_j^s - T_{i,j}^t\right) = \frac{\dot{m}}{n_t}\Delta h_t \quad (5)$$

For single phase flow, enthalpy is a function of temperature, which is found using a log-mean temperature difference method:[28]

$$\frac{T_i^s - T_{i+1,j}^t}{T_i^s - T_{i,j}^t} = \exp\left(-\frac{PL}{\dot{m}c_p}\overline{U}_{i,j}\right) \quad \text{for } j = 1,3,5 \quad (6)$$

$$\frac{T_i^s - T_{i,j}^t}{T_i^s - T_{i+1,j}^t} = \exp\left(-\frac{PL}{\dot{m}c_p}\overline{U}_{i,j}\right) \quad \text{for } j = 2,4 \quad (7)$$

where P is the perimeter and L is the element length. $\overline{U}$ is the average overall heat transfer coefficient:

$$\overline{U} = \left(\frac{1}{h_{in}} + \frac{r_{in}}{k}\ln\frac{r_{ex}}{r_{in}} + \frac{r_{in}}{r_{ex}}\frac{1}{h_{ex}}\right)^{-1} \quad (8)$$

The tube-side fluid is often of ambiguous phase, so equations of state for the thermodynamic variables are not useful. Instead, a nitrogen properties database is used, where the inputs are the pressure and either the bulk temperature or enthalpy, depending on the phase.[29] The external heat transfer coefficient is evaluated using the Churchill and Bernstein correlation for cylinders in crossflow.[30] The internal heat transfer coefficient for single phase flow is evaluated using Gnielinski's formula.[31] For two-phase flow, the heat transfer coefficient is found via Kandikar's correlation.[32]

Results

Results found for the design cruise mass flow of 60 g/s are given in Fig. 14. The shell-side temperature, as well as the tube-side quality as a function of the non-dimensional span are shown. The specific enthalpy at the economizer outlet is 92.3 kJ/kg, giving a heat transfer effectiveness of $\varepsilon_{econ} = 0.72$.

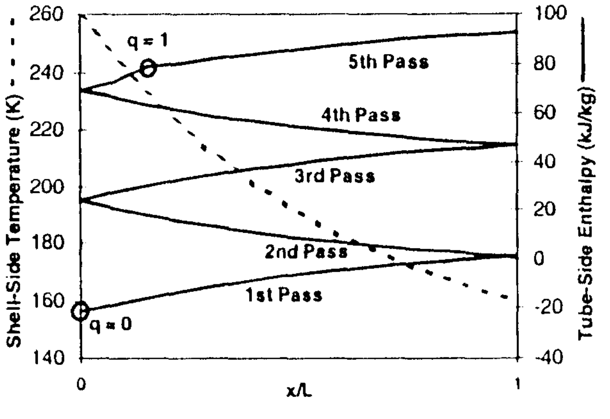

Fig. 14: Quality and temperature profiles through the economizer.

The thermodynamic performance, ε, of the economizer is inversely dependent on the mass flow. To examine why this occurs, note that for this heat exchanger, the shell-side resistance to heat transfer dominates. That is, for any element

$$\Delta h_t = \frac{Q}{\dot{m}} = \frac{T_s - T_t}{\dot{m}R_{tot}} \approx \frac{T_s - T_t}{\dot{m}R_s} = h_s A \frac{T_s - T_t}{\dot{m}} \qquad (9)$$

The external heat transfer coefficient scales as $\dot{m}^{0.6}$, so

$$\Delta h_t \propto \frac{h_s}{\dot{m}} \propto \frac{\dot{m}^{0.6}}{\dot{m}} = \dot{m}^{-0.4} \qquad (10)$$

This behavior is displayed by the solid line in Fig. 15. This shows the decreasing value of ε_{econ}, along with the resulting increase in the exit temperature of the shell-side fluid vs. mass flow. The variability in the curve is an artifact of the finite convergence criterion used in the heat transfer simulations and does not reflect any physical phenomena.

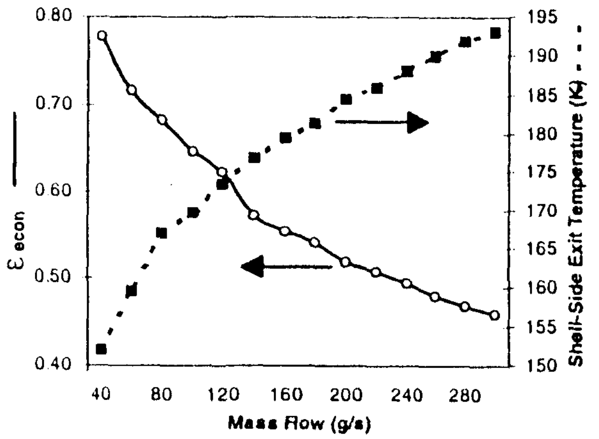

Fig. 15: Economizer effectiveness vs. mass flow.

The calculated performance of the economizer meets the original criteria of $\varepsilon_{econ} > 0.5$ for the majority of its operational envelope, but falls short of the requirement at high mass flows, which will typically occur only in transient bursts. Thermal transients for sub-ambient heat exchangers generally work in *favor* of performance, so a partial offset of this decrease in effectiveness is expected. To determine whether this is true requires extensive road testing, which was not possible as of this writing because construction on the test-bed vehicle was still in progress.

V. Ambient-Air Heat Exchanger Development

The goal of this research is to design a heat exchanger that can operate in a variety of environmental conditions and is structurally robust, while not being hampered by the build-up of frost. Many approaches have been examined,[14] but in general, strategies for dealing with frost formation fall into two categories: passive and active control.

Passive control involves either preventing frost formation, or oversizing the heat exchanger such that frost build-up is unimportant on time-scales characteristic of automotive travel. The advantages of passive control are mechanical simplicity and reliability. One disadvantage is that passive systems are, in general, less flexible in dealing with off-design operation.

Active control of frost formation entails allowing frost to accrete, and then removing it either thermally or mechanically. The advantage of active systems is that they are more operationally robust. Performance of an actively controlled heat exchanger is much less dependent on the ambient or operational conditions since these systems can be responsive to different frost loading conditions. Disadvantages include the power consumption and poor long-term reliability.

Both options have been examined closely. Of the two, the passive control heat exchanger was chosen for the *LN2000* as the best possible solution to the issue of frost formation. As of this writing, the ambient air heat exchanger has been built and pressure tested. The results presented in this chapter are calculated and have not yet been experimentally verified. Described herein is the theoretical modeling, design, and fabrication of the ambient-air heat exchanger. Future plans include rigorous testing to verify the calculations and identify possible modifications for a second-generation heat exchanger.

Design

The operating environment of a typical automobile can be a demanding one. There are weather conditions such as rain, snow, or extreme heat to contend with. There are driving conditions such as rough pavement, gravel roads, potholes and speed-bumps, and there are situations such as stop-and-go traffic, long-distance highway travel, and cold-starts. Finally, automobiles have to meet rigorous safety, comfort, and reliability standards if they are to compete in today's marketplace. A heat exchanger for automotive application has to successfully meet all of these requirements.

Constraints: Along with meeting the necessary design requirements, the automotive heat exchanger has to also fit within constrained volume and mass envelopes. The *LN2000* has a space in back, under the cargo deck, which provides a large volume and allows easy ducting of ambient air from beneath the vehicle. Furthermore, because the heat exchanger can be hung directly from the deck, structural modifications are minimal. The available space constrains the heat exchanger to dimensions of no more than 78 cm x 53 cm x 38 cm. The mass constraint is less well defined, but it does stand to reason that in an 1100 kg automobile with an 11 kW motor, less mass is better.

Configuration: To prevent frost buildup on subambient heat exchangers, the exterior surfaces must be kept above the freezing point of water. A method for achieving this is shown in Fig. 16. The principle is the same as for multi-fluid heat exchange systems, where heat is transferred from source to sink via a number of media operating in series. In this concept, that series of media is simply the nitrogen gas at different stages of its thermal history.

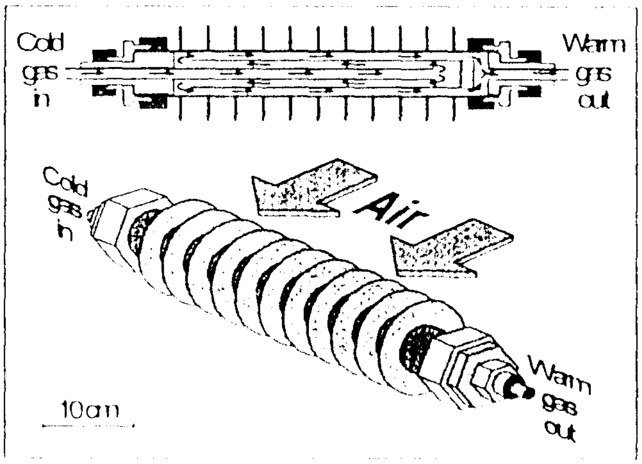

Fig. 16: Frost free heat exchanger element.

Figure 16 shows only three passes, but it is possible to use as few as two, or as many as can be fit within the outermost tube. Two concentric tubes, operating within a bundle in a shell-and-tube heat exchanger, are known as bayonets. These have been used for years to eliminated stresses due to differential thermal expansion between the shell and its tube bundle.[33] More recently, a 1995 patent for a frost-tolerant heat exchanger described much the same idea as that presented here, but using only two concentric tubes in a variety of configurations.[34]

The choice of three passes was based on two practical considerations. The first is that using an odd number of internal passages allows the inlet and outlet to be at opposite ends of the element, greatly simplifying the manifolding. The second is that going to more internal passages such as five or seven can present problems with fabrication, cost and mass. This has to be weighed against the fact that more passes increase the robustness of the system. Robustness insures that frost-free performance can be maintained in lower-temperature environments.

An example of the behavior of the gas as it passes through the heat exchanger element is given in Fig. 17. The black arrows represent heat transfer: from the ambient air to the wall, from the wall to the nitrogen and from each pass to the previous one. The exterior wall temperature is dependent on the heat transfer coefficient and fluid temperature in the *outermost* passage only.

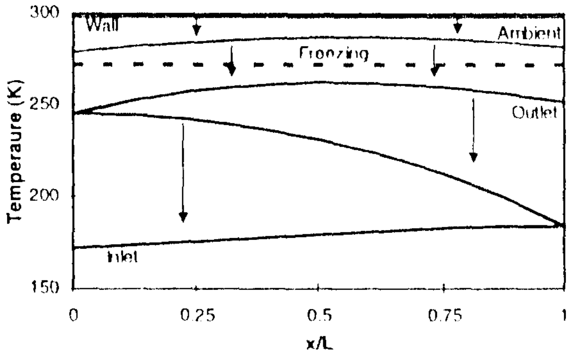

Fig. 17: Temperature profile of frost-free heat exchanger element.

Passive heat exchangers are more tightly coupled to environmental and operating conditions. In the case of this particular concept, when the outside temperature is lower than the freezing point of water, frost will form. In fact, frost will form even when the ambient temperature is above the freezing point of water, *if* the temperature difference from the air to the wall is too large.

This does not, however, mean that the heat exchanger has been rendered ineffective. Figure 18 shows the absolute humidity (kg water/kg air expressed as a percentage) in the atmosphere versus the ambient temperature on a day in which the relative humidity is 100%. This corresponds to the worst case frost formation scenario. It is evident that humidity is a very strong function of temperature. So much so that on cold days, though frost may form, there will be much less to inhibit heat transfer than when under more normal driving conditions. The threshold between frosting and non frosting conditions will be examined later in this section.

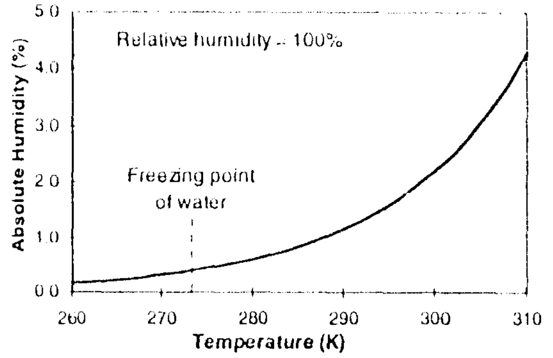

Fig. 18: Absolute humidity vs. temperature.

A more detailed view of the heat exchanger element is shown in Fig. 19. There are several notable aspects of the three-pass heat exchanger element which bear pointing out. The first is the use of a spring as a heat transfer augmentation device. The spring wire diameter is such that it very nearly occupies the entire space between the nested tubes. This provides a helical path along which the fluid can flow. This helical path creates a centripetal acceleration which enhances heat transfer.[15] It also produces a "fin" effect which provides a conduction path between the bulk fluid and the wall.[28] Finally, the springs provide a structural load path between the thin, inner tubes and the rigid outer one.

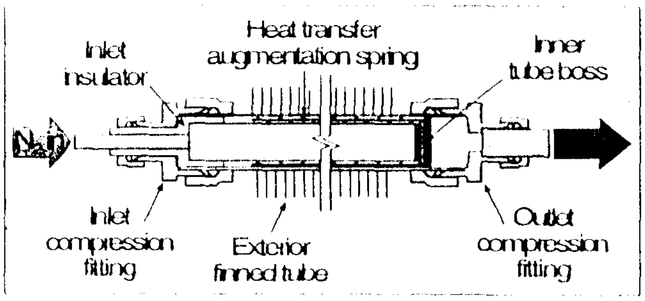

Fig. 19: Design aspects of the three-pass heat exchanger element.

Another innovation is the inlet insulator. This nylon plug provides a barrier to conduction between the cold inlet line and the outside wall. Without the insulator, heat would pass through the inlet compression fitting to such an extent that conduction would undermine the careful design of the convective heat flow pattern. The insulator also gives a natural attachment point for the inner tube structure, which otherwise would slide longitudinally within the external tube.

The final innovation in the three-pass heat exchanger element is the relative diameters of the tubes themselves. These particular diameters were chosen specifically to provide the temperature profile required to keep the outside wall above the freezing point of water.

As shown in Fig. 20, the ambient-air heat exchanger assembly consists of 45 elements, having a mass of approximately 109 kg. Thirty-nine of these elements form the primary heat exchanger and operate in parallel. Six of the elements are superheaters, also acting in parallel. Together, the primary and superheater sections of the heat exchanger operate in series. The superheaters are 51 cm sections of the finned tube without the internal ducting. These are intended to add a superheat beyond the bulk exit temperature of the 39 primary elements. This is because with the triple-pass concept, the exit temperature of each element is depressed below what is possible. As Fig. 17 shows, the temperature of the exiting gas drops from its peak value because of the heat transfer to the second passage. The superheater is added to regain some of that thermal energy. An additional advantage of the superheater is that it also provides a partial dehumidification of the incoming air before it reaches the colder, primary elements.

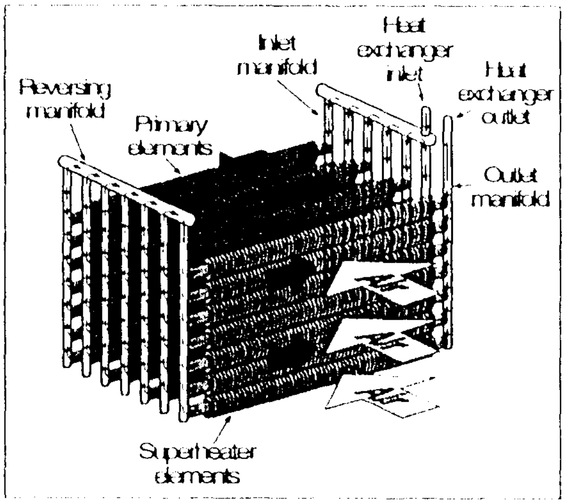

Fig. 20: Ambient-air heat exchanger flow pattern.

The air flow is channeled from under the vehicle and through the heat exchanger by a sheet metal intake. The air is propelled both by the motion of the van and by two adjustable speed 0-100 W ducted fans hung from the rear of the vehicle. Fan power for this test vehicle is provided by the main 12 VDC battery.

Materials and Fabrication: One of the most common techniques to increase heat transfer to tubes in crossflow is to apply extended surfaces, or fins. The outer tubing chosen was 25.4 mm OD copper tubing with a 12.7 mm integral helical fin wrapped at 275 fins/m. Each tube is 61 cm long with a finned length of 51 cm and has a mass of 1.6 kg. This choice drove the design of the rest of the element.

For performance reasons, the internal tubes were chosen primarily for their diameters - 17.5 mm and 20.6 mm - though wall thickness was also a factor. The internal tubing does not need to support a large pressure difference, so weight and fabrication issues drive the choice of wall thickness. This made them thin: 0.36 mm for the innermost tube and 0.81 mm for the second tube. Brass is the material of choice because of its ease of brazing and low cost. The two inner tubes are joined at their bases by a bronze boss, shown in Fig. 19. The joining was accomplished via a silver-solder brazing process.

The innermost tube is perforated at its base to allow the nitrogen to pass unimpeded into the second tube. This inner-tube structure is held in place with respect to the outer tube by a dowel pin set through both the innermost tube and the inlet insulator. All of the fittings are brass and are of the compression type. The springs were made of 0.74 mm mild steel wire.

The heat exchanger elements are attached to the manifolding with brass compression fittings. This was chosen to allow disassembly for repair. Each fitting has a mass of 0.34 kg, thus contributing a total mass of 30.6 kg.

The manifolding is constructed of 12.7 mm, 19.0 mm and 25.4 mm welded 304 stainless steel tubing. All the manifold fittings are of the compression type and are made of 316 stainless steel.

Structural Integrity: All of the manifold and finned tubing maintains at least a factor of safety of 2.5 at the system's operating pressure. The internal brass tubes, which see very little pressure differential, need not be designed for high pressure service. The weight of the heat exchanger assembly is supported by two plates which are bolted to the floor of the vehicle. Each of the heat exchanger's seven "racks" are hung from their inlet and outlet manifolds, which are clamped to the two plates. The racks transmit very little load through their pressure fittings into inlet and outlet headers.

Modeling

A model has been developed to simulate the heat transfer characteristics of the three-pass element. This has been an instrumental design tool for quickly evaluating changes in geometry and operating environments. For the sake of simplicity, many of the same assumptions have been made about the heat exchanger as were made about the economizer. Namely:

- Steady-state operation. Again, this is conservative since the thermal transients associated with startup improve performance.
- Even flow distribution between and within heat exchanger elements.
- Zero axial conduction.
- Zero radiation.
- Adiabatic corners.
- Radial symmetry.
- Single phase (gaseous), fully developed flow.
- Constant average heat transfer coefficients for each pass.

These assumptions allow the multi-pass heat exchanger to be modeled as is shown in Fig. 21. In this schematic, η is the non-dimensional span variable, defined as by x/L and Q_c is the heat transfer due to water condensation on the outside surface of the heat exchanger.

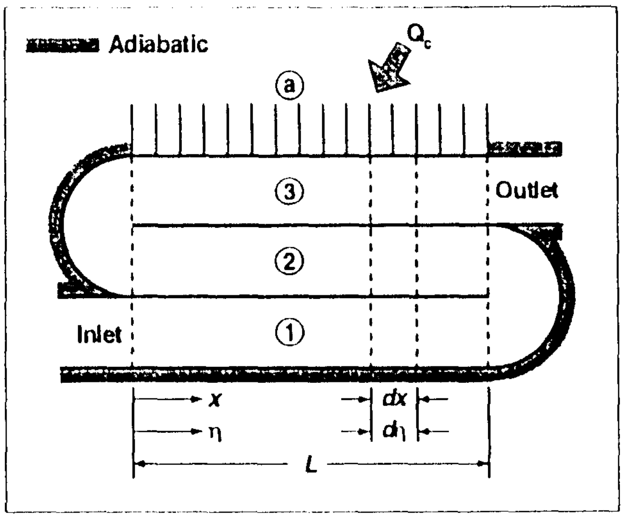

Fig. 21: Model of ambient-air heat exchanger element.

Applying the first law of thermodynamics to the differential element $d\eta$, the governing equations can be derived for the third, second and first passes, as shown in Eqs. 11-13:

$$-dQ_c + \dot{m}c_p dT_3 = P_3\overline{U}_{a3}(T_a - T_3)dx - P_2\overline{U}_{32}(T_3 - T_2)dx \quad (11)$$

$$-\dot{m}c_p dT_2 = P_2\overline{U}_{32}(T_3 - T_2)dx - P_1\overline{U}_{21}(T_2 - T_1)dx \quad (12)$$

$$\dot{m}c_p dT_1 = P_1\overline{U}_{21}(T_2 - T_1)dx \quad (13)$$

where P_i is the perimeter of the ith tube and $\overline{U}_{i,i-1}$ is the overall heat transfer coefficient between tubes i and i-1.

The governing equations can be nondimensionalized via the following definitions:

$$\tau_1 \equiv \frac{T_1}{T_a}, \qquad \tau_2 \equiv \frac{T_2}{T_a} \qquad \tau_3 \equiv \frac{T_3}{T_a} \tag{14-16}$$

$$K_1 \equiv \frac{P_1 L \overline{U}_{21}}{\dot{m} c_p}, \quad K_2 \equiv \frac{P_2 L \overline{U}_{32}}{\dot{m} c_p}, \quad K_3 \equiv \frac{P_3 L \overline{U}_{a3}}{\dot{m} c_p} \tag{17-19}$$

Q_c is calculated, assuming constant flux F, by

$$Q_c = \int_0^x F P_3 d\overline{x} = F P_3 x = F P_3 L \eta \tag{20}$$

which can be nondimensionalized by

$$K_4 = \frac{F P_3 L}{\dot{m} c_p T_a} \tag{21}$$

Substituting into the original equation, and writing in matrix notation, gives

$$\begin{Bmatrix} \dfrac{d\tau_1}{d\eta} \\ \dfrac{d\tau_2}{d\eta} \\ \dfrac{d\tau_3}{d\eta} \end{Bmatrix} = \begin{bmatrix} -K_1 & K_1 & 0 \\ -K_1 & K_1 + K_2 & -K_2 \\ 0 & K_2 & -(K_2 + K_3) \end{bmatrix} \begin{Bmatrix} \tau_1 \\ \tau_2 \\ \tau_3 \end{Bmatrix} + \begin{Bmatrix} 0 \\ 0 \\ K_3 + K_4 \end{Bmatrix} \tag{22}$$

The three boundary conditions for this problem are

$$\tau_1(0) = \tau_o \tag{23}$$

$$\tau_2(1) = \tau_1(1) \tag{24}$$

$$\tau_3(0) = \tau_2(0) \tag{25}$$

The behavior of this heat exchanger is determined entirely by the inlet temperature and the four nondimensional heat transfer coefficients, K_1, K_2, K_3 and K_4. These provide a convenient parameter against which the performance can be optimized to get above freezing wall temperatures. The K_i values are themselves complicated functions of the geometry and heat transfer coefficients, but they provide targets toward which the system design can be adjusted.

Unlike for the economizer, a closed-form solution for the thermodynamic performance of the heat exchanger has been developed. The heat transfer coefficients, however, are strongly dependent on such variables as the density and the temperature, which vary considerably from inlet to outlet. Therefore, an element-wise evaluation of the heat transfer coefficients, followed by a pass-by-pass averaging, must be performed to obtain the K matrix.

For the innermost passage, the flow correlation used was that given by Petukhov.[28,36] The annular heat transfer coefficients for the concave and convex walls were interpolated from the data given by Kays and Perkins.[37] The heat transfer augmentation factors provided by the spring inserts were calculated from the correlations of Kamath and Rao.[38] The external heat transfer coefficient was calculated using the equation of Churchill and Bernstein, modified with a fin efficiency parameter.[28, 30]

This algorithm can be wrapped in another iterative loop which calculates the performance of the entire heat exchanger assembly, including the superheater elements (which require another solution algorithm altogether).

Results

To verify the model of the three-pass heat exchanger element, an experiment was conducted in the wind-tunnel using the first fabricated element. The results, shown by the two circles in Fig. 22, are for a nitrogen mass flow of 1.78 g/s, an inlet temperature of 143 K, an ambient temperature of 297 K, an ambient humidity of 40% and an air speed of 5.1 m/s. The model underpredicts the exit temperature of the element by 6.1% and underpredicts the change in temperature (T_2 - T_1) by 14.2%.

This discrepancy may be due to the fact that for this simulation, the humidity was set to zero. This is despite the fact that enough moisture was condensed out of the air to form a puddle at the bottom of the wind tunnel after only 30 minutes of operation.. This reason this feature was not employed was because the solution would not converge if a non-zero humidity was entered. The model approximates the saturation pressure of water in the atmosphere with an exponential function of temperature. Because of the sensitivity of this function, the numerical iteration scheme would often become divergent after only two steps. For future work, it is recommended that more robust iteration procedures, such as Newton-Raphson or dynamic relaxation be used.[39]

During the experiment, frost formed over the first 6 cm of the inlet side of the element. As stated before, axial conduction is not modeled, so whether the formation of frost is due to convection from the flowing nitrogen, or conduction through the nylon plug and into the inlet fitting, is not known at this time.

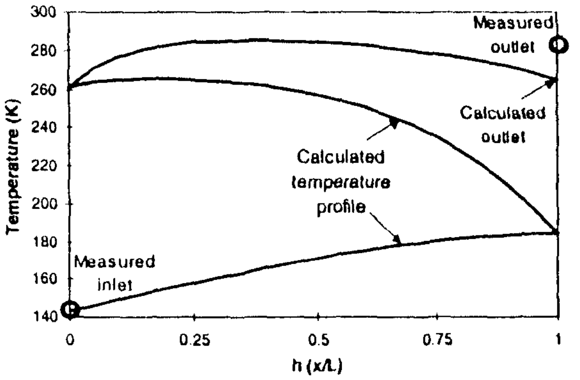

Fig. 22: Prediction of temperature profile for ambient-air heat exchanger element.

To predict the performance of the vehicle at cruise, this model was incorporated into an overall heat exchanger simulation. The results from the simulation of the economizer, for a mass flow of 60 g/s, can be used to establish the inlet conditions of the heat exchanger assembly. The ambient temperature is assumed to be 293 K, the air velocity is 5.1 m/s, and condensation is not considered. The results of this simulation are given in Fig. 23. The calculated exit temperature of the superheater (and therefore of the entire heat exchanger) is 269 K for an inlet temperature of 126 K. The air-side exit temperature is 285 K, giving a temperature change of 8 K.

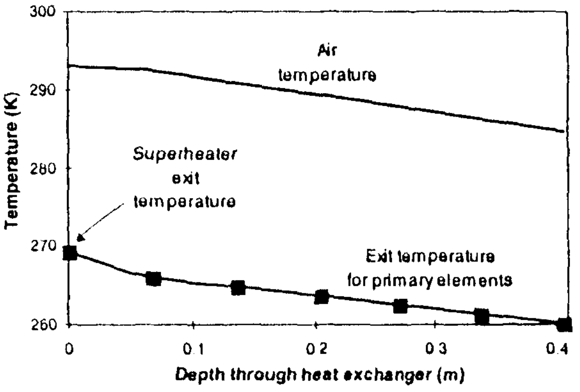

Fig. 23: Temperature through ambient-air heat exchanger at cruise.

At higher mass flows, calculated performance begins to decrease. Figure 24 shows both the air-side and tube-side exit temperature of the ambient-air heat exchanger as a function of nitrogen mass flow. This assumes a constant inlet temperature of 126 K, a wind speed of 5.1 m/s and an ambient temperature of 293 K. Clearly, thermal performance degrades with increasing mass flow. Moreover, the air-side exit temperature drops below the freezing point of water, guaranteeing at least some frost formation.

Setting a constant inlet temperature for increasing mass flow provides an apples-to-apples comparison, but is somewhat artificial. Because the heat exchanger model does not account for two-phase flow, an inlet temperature slightly higher than saturation must be employed (i.e., 126 K at 34 bar). However, as was shown in Fig. 15, the tube-side exit temperature of the economizer decreases with increasing mass flow. Above about 100 g/s, the exiting fluid is saturated. Therefore, an overall heat exchange system simulation would predict results lower still that those shown in Fig. 24.

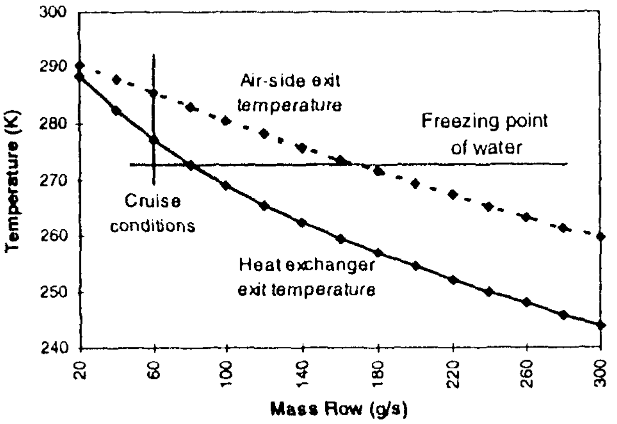

Fig. 24: Ambient-air heat exchanger outlet temperature vs. mass flow.

To examine the sensitivity of performance to changes in air speed, net power output (shaft power minus fan power) vs. fan air speed is shown in Fig 25. This assumes that the motion of the vehicle does not induce air flow, as would be the case for low-speed acceleration. Clearly, there is an optimal fan air speed for different mass flow rates of nitrogen. This goes up as the mass flow of nitrogen increases. Higher air speed also improves the resistance to frost formation by decreasing the air-side temperature drop through the heat exchanger. While increasing the fan-air speed does ensure peak power output for a given mass flow regime, it also reduces the specific power, lowering the overall efficiency. It is recommended that a feedback control system for the fan motors be incorporated into future designs to optimize performance.

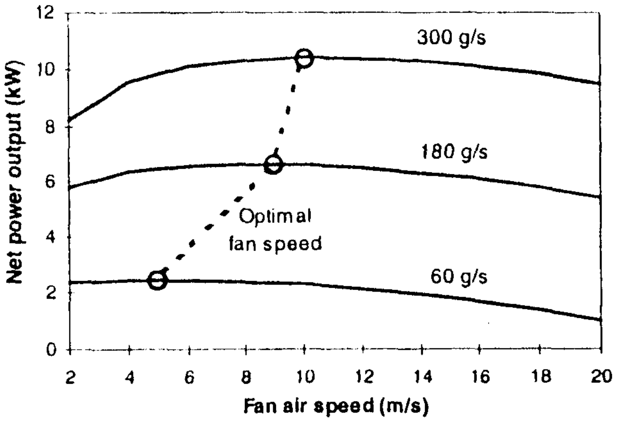

Fig. 25: Net power output vs. fan air speed.

This type of heat exchange system operates most efficiently, and with the least frost-formation, at mass flows corresponding to cruise conditions. For a cryogenic automobile with a hybrid-electric drivetrain, that is exactly what is required. A hybrid-electric design would allow the heat exchange system to run steady-state, while a battery pack or other energy storage device provided load leveling. It is therefore recommended that future designers evaluate the drivetrain selection with an eye toward the thermal performance of the vaporization system.

VI. Conclusions

The cryogenic automobile is a potential contender in the ZEV market, provided certain key technologies are demonstrated. One of these technologies is the development of an all-weather heat exchange system. A heat exchanger that works well in a variety of operating conditions, is structurally robust, and is not hampered by the buildup of frost is a necessary technology for the cryogenic automobile. This paper describes the design and fabrication of such a system. Although experiments have yet to be conducted to determine the actual performance, numerical predictions have been given.

The first stage of the heat exchange system is the economizer, which is a shell-and-tube heat exchanger that uses the exhaust gas from the expander to preheat the incoming liquid nitrogen. Performance simulations indicate that the design presented here is adequate for the calculated cruise mass flow. At higher mass flows, however, performance degrades to the point that complete vaporization of the nitrogen is not possible. The shell-and-tube configuration of the economizer appears to be adequate for the immediate application, but it is recommended that for future designs the tube density be increased.

The second stage of the heat exchange system is the ambient-air heat exchanger. The elements of this unit employ a triple-pass configuration to maintain the external wall temperature above the freezing point of water. Simulations show that, like the economizer, performance is adequate at cruise conditions, but degrades with increasing mass flow. Unlike for the economizer, this can be (at least partially) offset by increasing the fan power and therefore the air velocity and mass flow through the heat exchanger core. This not only increases the heat transfer to the nitrogen but also diminishes the likelihood of frost formation by minimizing the air-side temperature drop. It is recommended that an automatic fan-speed control system be incorporated into future designs.

The ambient-air heat exchanger is currently too heavy to satisfy modern automotive design constraints. This design would be improved with the use of aluminum finned tubes rather than copper. Similarly, because the internal ducts see very little pressure differential, these could be made of plastic instead of brass. The compression fittings also carry a significant weight penalty, so welded or brazed joints would be preferable.

Two recommendations for future modeling efforts are to include condensation of ambient humidity and to incorporate transient analysis. Including condensation is a matter of utilizing a more robust iteration scheme. The incorporation of transient analysis entails adding another layer to the nested iteration algorithm.

Finally, the numerical simulation of the heat exchange system has allowed insight into its behavior and has provided a useful design tool. But, until road-test experiments are conducted, the actual thermal performance will not be known with enough accuracy to support an overall recommendation for the final heat exchange system design.

Acknowledgments

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Quasi-Isothermal Expansion Engines for Liquid Nitrogen Automotive Propulsion

C. Knowlen, J. Williams, A.T. Mattick, H. Deparis, and A. Hertzberg
Aerospace and Energetics Research Program
University of Washington
Seattle, WA

ABSTRACT

An automotive propulsion concept is presented which utilizes liquid nitrogen as the working fluid for an open Rankine cycle. Ambient heat exchangers are used to power an engine that is configured to maximize heat transfer during the expansion stroke. If sufficient heat input during the expansion process can be realized then this cryogenic propulsive system would provide greater automotive ranges and lower operating costs than those of electric vehicles currently being considered for mass production. The feasibility of meeting this engineering challenge has been evaluated and several means of achieving quasi-isothermal expansion are discussed.

INTRODUCTION

The potential of cryogenic energy storage for automotive propulsion has been under investigation at the University of Washington as an alternative to electrochemical batteries for zero-emission vehicles (ZEV). It is anticipated that the use of an inert cryogen, such as liquid nitrogen (LN_2), as an energy storage medium would not pose any environmental burden, and in particular would avoid the issues of heavy metals pollution associated with lead-acid batteries. Another potential advantage over electro-chemical batteries is that the installation of a cryogen distribution system could be realized by straight-forward modifications to the existing petroleum retail stations. Several means of utilizing the thermal energy potential of cryogens have been examined and their performance capabilities are presented.

The basic idea of the LN_2 propulsion system is to utilize the atmosphere as a heat source to vaporize and superheat the cryogenic fluid for use in a thermal power cycle. This is in contrast with typical heat engines which utilize an energy source at temperature significantly above ambient and use the atmosphere as a heat sink. In both cases the efficiency of conversion of thermal energy of the source to work (W) is limited by the Carnot efficiency, $\eta = W/Q_h = 1 - T_l/T_h$, where Q_h is heat input, T_l is the sink temperature, and T_h is the temperature of the heat source. The ideal thermal efficiency is impressively high (74%) for a heat engine operating on the temperature difference between ambient ($T_h = 300$ K) and LN_2 stored at atmospheric pressure ($T_l = 77$ K). The ideal work recoverable from an expansion engine as LN_2 is evaporated and brought up to ambient temperature is given by the difference in thermodynamic availability between liquid and ambient states. This "reversible work" is $W_r = 769$ kJ/kg-LN_2 which is significantly higher than the 180-300 kJ/kg achieved with lead-acid batteries, thus only 40% of the reversible work needs to be recovered to provide the LN_2-propelled vehicle with a driving range commensurate with that of advanced battery-powered vehicles of comparable weight. The key issues are the ability to design a practical energy conversion system that can take advantage of this high thermal efficiency and the available energy of the cryogen while still being cost competitive with conventional electric vehicles.

BACKGROUND

Prior work conducted at the University of Washington has indicated that significant gains in the overall energy efficiency of an ambient heated LN_2 propulsion system would result if an effective isothermal engine could be developed.[1] Figure 1 shows the specific work plotted as a function of peak cycle pressure, with peak temperature as a parameter, for ideal adiabatic and isothermal expanders (LN_2 pump efficiency of 80% is assumed). The cut-off point for LN_2 injection is adjusted so that the final pressure is always 0.11 MPa. For the range of injection pressures shown the isothermal work increases monotonically with increasing pressure, whereas the adiabatic cycle is only weakly dependent on peak cycle pressure above 4 MPa. The inclusion of pump work results in a specific work maximum for both expansion processes, even though it is not evident in the isothermal work plotted here. It is apparent that the need to efficiently use LN_2 dictates that the expander operate at as high a pressure as mechanically feasible and that heat transfer be maximized during the expansion process.

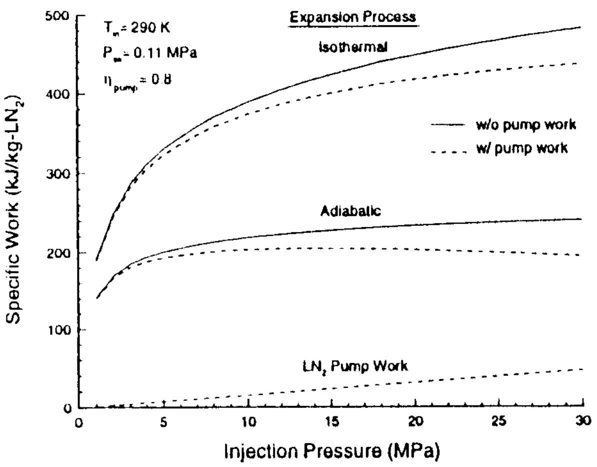

Fig. 1 Specific work output for ideal isothermal and adiabatic expansion processes.

REVIEW OF PATENTS The potential for using liquefied air and LN_2 for vehicle propulsion has been disclosed in several patents issued since the early seventies. Boese and Hencey [1972] proposed a hybrid system in which the cryogen was pressurized and ambient heated to drive a gas motor which turned an alternator, the output from which is used to power an electric motor whose efficiency is enhanced with LN_2 circulation.[2] A second patent was issued to Boese in 1981 for a LN_2 system that scavenged heat from the passenger compartment for air conditioning in conjunction with ambient heat exchangers and expelled the pressurized gas through a rotary vane type turbine.[3] The suggestion of a "master expander coil", i.e., a gas receiver, was also introduced for this system. Manning and Schneider [1974] patented a direct drive system utilizing multiple expansions with intermediate reheats followed by a final stage of gaseous recompression and subsequent heating prior to expansion (Brayton cycle).[4] They even went so far as to propose a regenerative device which alternatively routed cold N_2 exhaust and warm ambient air through a volume which contained many tension wires connected to a piston that contracted and relaxed under the thermal cycling to extract the last bit of available energy from the working fluid.

The implementation of LN_2 fueled vehicles by fleets, metropolitan buses, and even golf courses is discussed by West et al. [1976] in their patent which focuses on the details of a double acting piston expander to efficiently use the working fluid.[5] Oxley [1980] suggests the liquefaction and separation of air as a means of load-leveling for conventional power plants.[6] The oxygen can be used to enhance the efficiency of fuel combustion and the liquefied gases can be boiled by circulating them around the low temperature end of a Stirling engine using helium as a working fluid. In all of the works mentioned above the expansion processes were considered to be adiabatic and reheats with multiple expansions were invoked to utilize the cryogen fuel as efficiently as possible.

In a patent issued to Latter et al. in 1982, a system was designed to improve the mileage from conventional fuels (diesel, LNG, etc.) by operating a Rankine cycle which converted as much work as practical from an ambient-heated cryogen working fluid (LN_2), utilizing a series of reheats before topping the cycle with heat addition from a combustor.[7] A variation of this scheme was to use liquid air first as the working fluid of the Rankine cycle before injecting it with additional fuel into an internal combustion engine. Combining ambient and combustion heat in these systems enabled the range of the vehicle to be increased by 50% over that of using each fuel source separately. Scavenging heat from engine friction and vehicle braking was also proposed.

Even though many researchers have investigated the energy storage potential of cryogens and have developed several means for converting it to useful work, they apparently did not consider the potential for achieving isothermal expansion with a "cold" engine to be very promising. While this poses a non-trivial engineering challenge for high pressure expanders, a significant potential exists because the engine is operating in an environment that is at the peak temperature of the thermal cycle. Thus a study of the possibility of developing a quasi-isothermal expansion engine has been initiated and the current results are presented in this paper. The modifier "quasi" has been added to acknowledge that finite temperature differences and heat transfer rates will ultimately limit the performance of such an engine.

LIQUID NITROGEN PROPULSION SYSTEM

A schematic of the proposed LN_2 propulsion system is shown in Fig. 2. The cryogen "fuel" is stored in a vacuum jacketed vessel which would have appropriate relief features to safely accommodate boil off. A cryopump would pressurize the fluid to the supercritical operating pressure of

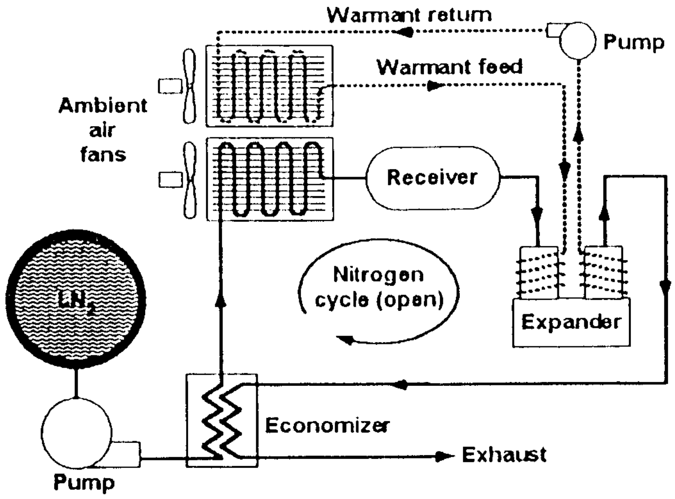

Fig. 2 Schematic of liquid nitrogen propulsion system.

the system and a carefully designed heat exchanger will enable the working fluid to be warmed to near ambient temperature without suffering a detrimental build-up of frost. The gaseous N_2 fills a receiver which minimizes pressure surges in the system due to changing power demands. Variable timed valves provide controllable cut-off to release N_2 gas into the expander as needed. More details of these components can be found in the cited references.[8,9]

A "warmant" fluid is circulated through the expander walls to maintain them at near ambient temperature (Fig. 2). The warmant must be pumped through another heat exchanger system (a conventional radiator would suffice) to efficiently conduct ambient heat into the engine. A quasi-isothermal reciprocating expander is proposed for this embodiment and its work output is transmitted to the wheels by means of a conventional transmission. Since the exhaust N_2 is near ambient temperature it is prudent to use it to preheat the pressurized LN_2 in an economizer to minimize contact of cold cryogen lines with ambient air.

THEORETICAL ANALYSIS OF EXPANDER

The thermodynamic simulation of a reciprocating expander has been developed to examine the impact of various engine design and operational parameters on the LN_2 consumption of an ambient powered cryomobile. A theoretical model for heat transfer was used in this analysis to determine the effects of injection pressure and temperature on the work output of the expander as a function of bore, stroke, and engine RPM.[10] The simulation procedure was generalized to enable the examination of a wide range of piston-cylinder head configurations. The warmant flow rate and ambient heat exchanger size were determined for the most promising configuration studied here. The details of the simulation methodology are discussed in this section.

RECIPROCATING ENGINE MODEL A one-zone, time-dependent analysis is applied on the control volume bounded by the piston and cylinder walls shown in Fig. 3. The cylinder contents are assumed to have uniform properties and be governed by the ideal gas equation of state. The heat transfer to the gas is coupled with the expansion process once the cut-off point is reached and ceases when the piston reaches bottom-dead-center (BDC). The surface temperature of all internal surfaces is assumed to be constant. The work output is then determined from the pressure-volume indicator diagram.

The instantaneous state of the N2 gas in the control volume is determined from energy conservation and the ideal gas equation of state. The energy equation for the cylinder contents is expressed as:

$$dP/d\theta = (\gamma-1) / V \{dQ/d\theta - \gamma / (\gamma-1)\, P\, dV/d\theta + dm_i/dt\, h_i - dm_e/dt\, h_e\} \qquad \text{(Eq. 1)}$$

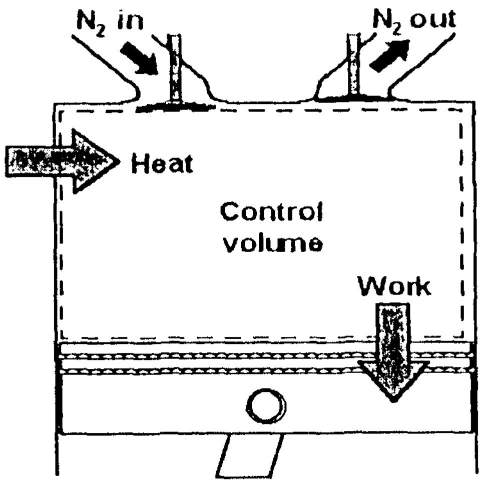

Fig. 3 Control volume for expander analysis.

where P is pressure, θ is crank angle, γ is specific heat ratio, V is volume, Q is heat transfer, dm/dt is mass flow rate, h is enthalpy and the subscripts i and e refer to injection and exhaust gas properties, respectively. The cylinder volume is obtained from the standard slider-crank relationship as follows:

$$V = V_h + V_c + \pi d^2 S / 8\, (1 - \cos\theta + R - (R^2 - \sin^2\theta)^{1/2}) \qquad \text{(Eq. 2)}$$

where d is the cylinder diameter, S is the length of stroke, and R is the ratio of connecting rod to crank radius. The clearance volume V_c is a measure of the minimum gap between piston and cylinder head; whereas the head volume V_h accounts for all the volume inside the expansion chamber that is not occluded by the piston when it is at top-dead-center if $V_c = 0$. The expression for $dV/d\theta$ is found by differentiating Eq. 2.

PISTON-CYLINDER HEAT TRANSFER In order to evaluate the trade off between bore, stroke, and revolution rate of a practical reciprocating expander, a means for estimating the heat transfer rate to the N_2 during the expansion process is required. Much of the piston-cylinder heat transfer research in the open literature has been directed to examining heat losses in fired engines and the resulting empirical correlations for average heat transfer coefficient are affected by radiative heat transfer. "Motoring" studies which determine the heat losses from an engine without combustion are closer to the desired situation and provide experimental data to validate the proposed heat transfer model. For purposes of this study the heat transfer from the cylinder walls is assumed to be similar to turbulent heating of gas in a tube as follows:

$$Nu_d \sim Re_d^{\,m}\, Pr^{n} \qquad \text{(Eq. 3)}$$

where Nu is the Nusselt number, Re is Reynold's number, and Pr is Prandtl number. The m exponent is typically assumed to be 0.8 for fully developed turbulent flow and n = 0.3 or 0.4

for cooling or heating, respectively. The functional dependences on temperature for the N_2 thermal conductivity k and viscosity μ in the temperature range of 200 - 300 K were determined from examination of thermodynamic data tables to be; $k \sim T^{0.85}$ and $\mu \sim T^{0.76}$, respectively.

Assuming that the gas velocity is proportional to the average piston speed U_p results in the heat transfer coefficient h_x having the following dependence on engine parameters:

$$h_x = (0.1129)\, d^{0.2}\, P^{0.8}\, U_p^{0.8}\, T^{0.594} \qquad \text{(Eq. 4)}$$

where the constant was determined for SI units by matching the heat transfer coefficient to an experimental data point found in Ref 11. The effects of piston stroke S and engine revolution rate N are incorporated in the average piston speed. This approach is not meant to rigorously quantify the heat transfer but rather to enable a reasonable investigation of the performance capabilities of various engine configurations.

The rate of heat input to the gas is the determined by assuming that h_x applies to all surfaces in the expansion chamber and that the wall temperature T_w is uniform and constant throughout the cycle. The corresponding heat transfer is:

$$dQ/dt = h_x\, A_w\, (T_w - T) \qquad \text{(Eq. 5)}$$

The total surface area exposed to heat transfer, A_w, at any crank angle is determined from:

$$A_w = A_p + A_h + A_c + \pi\, d\, S / 2\, (1 - \cos\theta + R - (R^2 - \sin^2\theta)^{1/2}) \qquad \text{(Eq. 6)}$$

where A_p is area of piston face, A_h is internal area of head volume, and A_c is surface area of clearance volume that is always exposed.

PISTON RING FRICTION The lost work due to piston ring friction is accounted for since it is expected to be highly dependent on the cylinder aspect ratio. The gas pressure is assumed to expand the rings against the cylinder walls and a constant value for the sliding friction coefficient (μ_r) corresponding to the "equivalent coefficient of friction under non-hydrodynamic conditions" is used.[12] Thus the differential element of friction work δW_f for the compression ring can be expressed as:

$$\delta W_f = \mu_r\, P\, \pi\, d\, t_r\, \delta S \qquad \text{(Eq. 7)}$$

where t_r is the ring thickness and δS is fraction of piston stroke through which this force acts. This expression is integrated over a complete engine cycle to account for the work lost to friction, which is then subtracted from the net cycle work. This energy is primarily dissipated into the cylinder walls and piston rings; however, its influence on the heat transfer is ignored at this time.

WARMANT CIRCULATION SYSTEM The warmant fluid is assumed to be a standard antifreeze mixture of water and ethylene-glycol. A conventional automobile radiator and water pump are proposed for this application since they are readily available. The water pump is assumed to have a 90% mechanical efficiency. The ambient heat transfer takes place at a lower temperature difference than in conventional automobiles; however, the amount of heat input required is less than that typically needed to be rejected from the fuel-burning engine. Heating of warmant in the ambient heat exchanger is governed by turbulent duct flow (Eq. 3) and the friction factor f is determined from Reynold's analogy as follows:

$$f / 8 = St\, Pr^{2/3} \qquad \text{(Eq. 8)}$$

where St is the Stanton number. The heat transfer coefficients for the warmant flowing around the expansion chamber and through the head based are the heat transfer correlations for cross-flow over cylinder walls and that through a flat plate for the cylinder head as follows:[13]

$$Nu_c = (0.35 + 0.56\, Re_d^{0.52})\, Pr^{0.3} \qquad \text{(Eq. 9)}$$

$$Nu_h = 0.332\, Re_d^{1/2}\, Pr^{1/3} \qquad \text{(Eq. 10)}$$

An energy balance for a given fluid temperature change determines the warmant flow rate needed to provide the desired cylinder heat flux through the expansion chamber walls.

ANALYTICAL PROCEDURE The analysis proceeds by first computing the temperature and pressure history for a fixed mass of gas undergoing expansion in a cylinder having a uniform wall temperature that is fixed at some point below ambient. The injection gas temperature is always equal to the wall temperature and the energy equation (Eq. 1) is integrated by the fourth-order Runga-Kutta method. The indicated work of the pressure-volume diagrams for the adiabatic and isothermal cycles are compared to the corresponding analytical values to insure sufficiently small steps are used for the integration. The effects of changing gas temperature and pressure on the heat transfer coefficient are accounted for during the expansion stroke by means of Eq. 4, using the mean piston speed. The work of the LN_2 and water pumps and the losses due to piston friction are independently calculated.

The intake valve is opened when the piston is at top-dead-center (TDC) and closed at the cut-off angle corresponding to expansion to desired exhaust pressure. This initial volume prior to expansion is analytically determined from the input pressure ratio for the adiabatic and isothermal cases, and iteratively determined in the case of heat transfer. When the initial volume is smaller than the sum of the head and clearance volumes the gas is assumed to instantly fill these volumes to the desired injection pressure and the expansion begins from TDC. This results in a higher exhaust pressure than initially specified. No heat transfer is assumed during the injection and exhaust phases of the cycle for the performance calculations presented in this paper.

RESULTS

The first set of calculations investigated the effects of bore-to-stroke ratio (*d/S*) and engine RPM on the isothermicity (*I*) of the cycle which is defined as the ratio of cycle work w_{cyc} to the isothermal work w_i determined for the same pressure and volume ratios. The isothermicity of a set of engines having a 1 cm bore diameter operating over an injection pressure range of 1 - 30 MPa with an exhaust pressure of 0.11 MPa and injection temperature of 290 K is shown in Fig. 4. Engine RPM and *d/S* are used as parameters. It can be seen that very long strokes and low RPM encourage heat transfer during the expansion process. The case of a 10 cm piston having a 1 : 4 bore-to-stroke ratio is also shown in Fig. 4. The reduction in work output due to bore size is evident. These calculations indicate that small bore and low RPM are desirable for an efficient quasi-isothermal engine. The corresponding low power density will result in careful design trade-off between high efficiency and the potential mechanical complexity of multiple-cylinder engines required to generate sufficient power for automotive propulsion.

The temperature and specific work as function of crank angle for an engine having a 1 cm diameter piston and 10 cm stroke operating at 1000 RPM under isothermal and non-adiabatic conditions are shown in Fig. 5. The cut-off point is between 12° and 13° for these conditions. It can be seen that the majority of the lost work in the non-isothermal case arises in the first 60° of crank angle. During this phase of the expansion process the highest rate of work output is realized, thus creating the greatest demand for heat input to approach isothermal performance.

The temperature difference between wall and gas is initially zero during the injection phase of the cycle; however, it quickly increases as the system does work on the piston, as shown in Fig. 6. The heat transfer coefficient rapidly decreases as the pressure drops, even though the increase in piston speed and decrease in gas temperature mitigate this effect. Since a well designed quasi-isothermal expander would have minimal temperature difference between wall and gas, the primary means of enhancing heat transfer are to increase both the heat transfer coefficient and total wetted area within the expansion chamber.

In order to determine the possible benefits of an enhanced heat transfer coefficient, the simulation for the non-isothermal expander was conducted again assuming that h_x did not decrease during expansion. The specific work output and temperature during the first half of the expansion stroke for the same operating conditions, when h_x is assumed to maintain its initial value, are shown in Fig. 7 along with the isothermal results. The work and temperature profiles are much closer to the ideal in this case. These results correspond to increasing h_x by a factor of 10 during the expansion over the first 60° of crank angle. The ratio of cycle work to isothermal is 0.85 under these conditions as compared to 0.65 for the variable h_x case, which is a significant increase in isothermicity. Increases in the heat transfer coefficient can be realized by inducing turbulence and swirl during the injection process, however, this approach has not yet been pursued sufficiently to evaluate the its feasibility of achieving the needed magnitude of h_x enhancement.

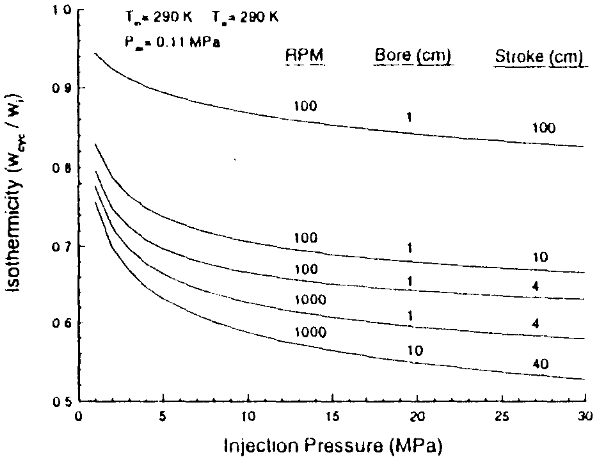

Fig. 4 **Effect of geometry and engine RPM on the isothermicity of an ideal expander with finite heat transfer rates.**

Since the baseline piston-cylinder configuration used for the prior simulations has a low isothermicity even though it has a very high surface-to-volume ratio (1 cm bore x 10 cm stroke) as compared to conventional expanders, something much different is required to achieve significant quasi-isothermal operation. One possibility is a novel piston-head

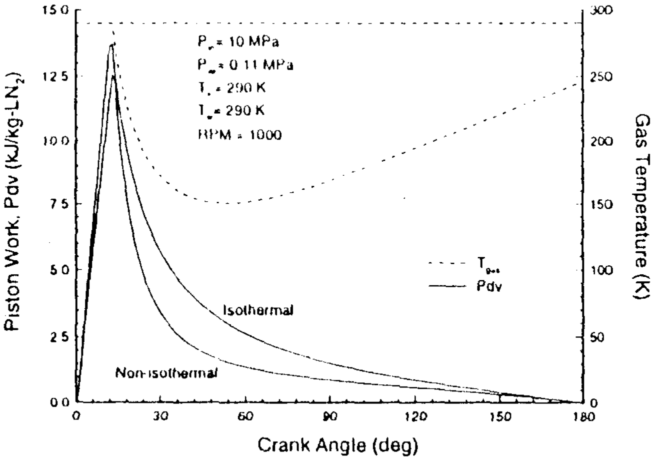

Fig. 5 **Instantaneous work output and gas temperature as function of crank angle for ideal expander. (Piston diameter = 1 cm, stroke = 10 cm.)**

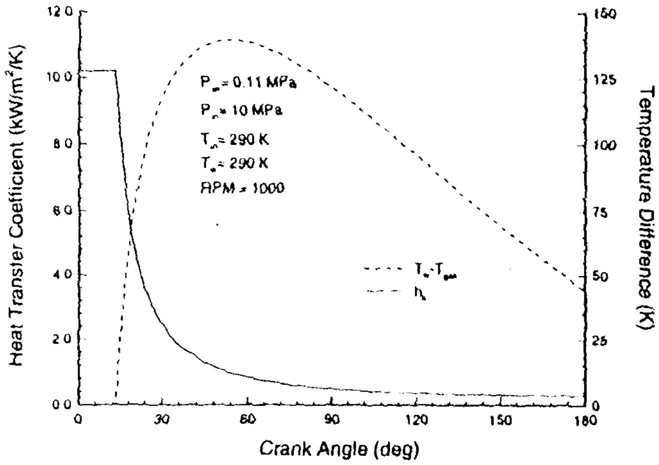

Fig. 6 **Heat transfer coefficient and temperature difference between gas and wall as function of crank angle.**

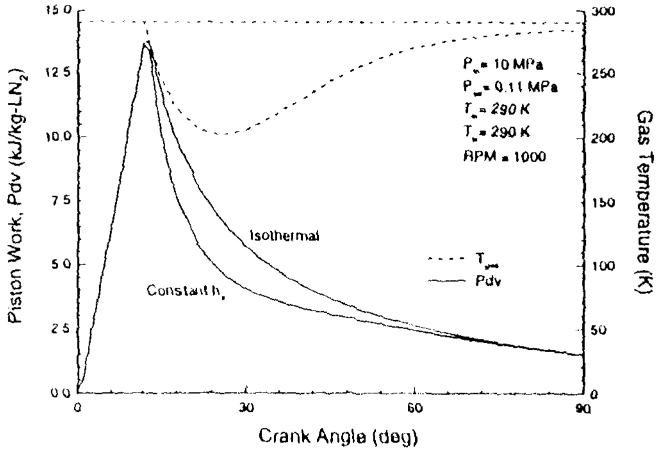

Fig. 7 **Instantaneous work output and gas temperature for ideal expander when h_x is assumed constant. (Piston diameter = 1 cm, stroke = 10 cm.)**

configuration that has the promise of greatly enhancing the heat transfer rate during the first part of the expansion process. This concept consists of a piston with multiple conical fins that fit into a heater core, which is imbedded in the top of the expansion cylinder, when the piston is at TDC, as shown in Fig. 8. The baseline configuration for this analysis consists of a 10-cm diameter piston fitted with 19 truncated conical fins having root and tip diameters of 1.45 and 0.725 cm, respectively. The fin height is 2.9 cm and the wall thickness of the core passages is 2 mm. Two plates of 0.5 cm thickness hold the core tubes and create a means for warmant to be routed over the tube bundle.

A head space over the heater core is, used for minimizing the pressure drop associated with distributing the N_2 through the passages. The pressure drop through the heater core is assumed to be negligible for this analysis.

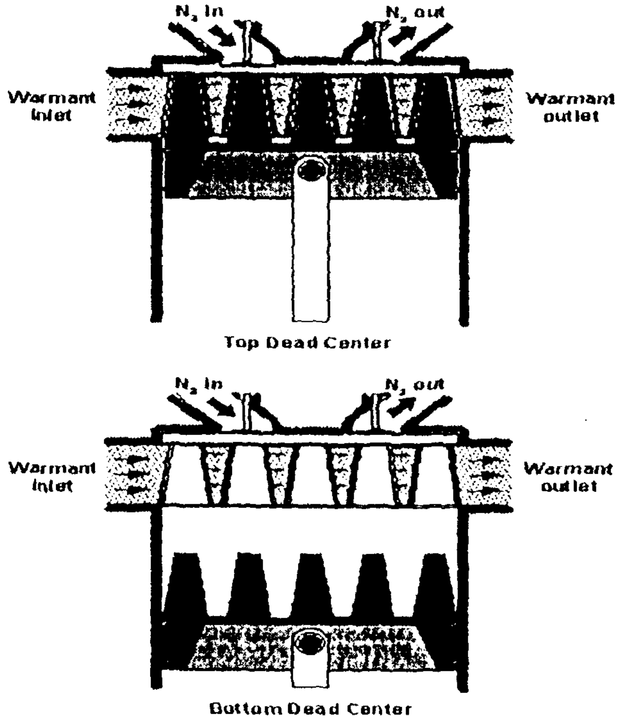

Fig. 8 **Piston having conical fins and heater core inside expansion cylinder.**

When the injection valve is opened at TDC, the gas experiences more surface area as the conical fins are withdrawn from the heater core. No seal is required between the fins and passages; the rings on the cylindrical portion of the piston fulfill this function. Thus a high surface-to-volume ratio is attained at the top of the cylinder where the demand for heat input is the greatest.

The work output for this piston-cylinder combination without any clearance volume or head space is shown in Fig. 9 along with that of a similar piston without fins. The

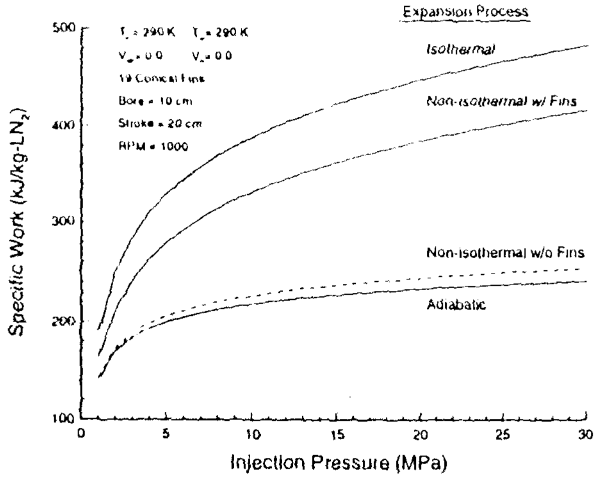

Fig. 9 **Work output from ideal expander using conical-finned piston and heater core without any clearance volume or head space.**

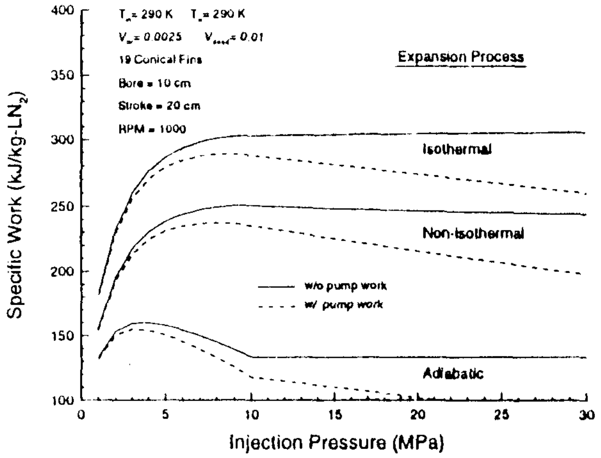

Fig. 10 **Work output from ideal expander using conical-finned piston and heater core with finite clearance and head volume.**

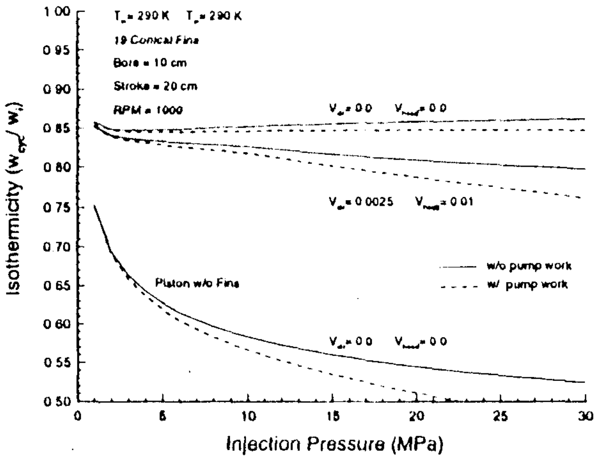

Fig. 11 **Isothermicity of conical-finned piston with and without clearance and head volume compared with standard piston.**

heat transfer is significantly enhanced beyond that of the finless configuration as evident in the difference in specific work curves. Without fins, a piston-cylinder configuration of these dimensions would be nearly adiabatic. The improvement in specific work is encouraging because the dimensions of this system are comparable to those of conventional expanders.

The work output of the finned-piston when there is finite clearance volume and head space is shown in Fig. 10. It can be seen that a significant penalty is incurred when these volumes are included. Adding in the pump work shows that the maximum specific work occurs in the 6 - 8 MPa range, which is well within the capabilities of current expander technology. The kink in the adiabatic specific work curve indicates the point where the high pressure N_2 could not be expanded to ambient (0.11 MPa) with the available expansion ratio, thus the exhaust pressure steadily increases with injection pressure and the pressure ratio remains constant. Even if the pressurization of the system is achieved without a pump, there is not any efficiency to gain by going to pressures higher than 10 MPa for any of these expansion processes.

The degree of isothermicity achieved with the finned-piston, with and without head space and clearance volume, is shown in Fig. 11 along with the isothermicity of a finless piston for comparison purposes. These calculations indicate that over 80% of the isothermal work available can be generated by the proposed expansion engine. This degree of isothermicity is nearly that which was previously found if h_x could be maintained constant during the expansion process. It can be seen that the presence of dead volume not only reduces the available work, it also reduces the isothermicity of the expansion process at higher pressure. This effect is also evident in the isothermicity curve for a standard piston. Thus careful piston-head design is essential to realize the full potential of using cryogens as energy storage media. Even with the specific work degradation due to finite head space, the performance of this engine is still good enough to be effectively used in a liquid nitrogen vehicle.

WARMANT CIRCULATION

The warmant circulation system is sized such that it can deliver the total heat input required to maintain the wall temperature of a 2-cylinder expander at 10 K below ambient when it is generating twice the power required for level cruising at 97 km/hr. This insures that there will be sufficient heat input during nominal cruising conditions to quickly overcome transient temperature fluctuations arising from acceleration and variations in road inclination. The power required to maintain cruising speed for a Honda CRX has been estimated at 7.8 kW.[14] This can be generated by a single cylinder of the finned-piston engine when it is operating at 850 RPM with an injection pressure and temperature of 6.0 Mpa and 290 K, respectively, in an 300 K environment. The heat input to the cylinder is ~11 kW under these conditions. Thus the maximum warmant heat transmission requirement is 22 kW which dictates a mass flow rate of 1.1 kg/sec (water/ethylene-glycol mixture) for a 5 K temperature drop. Under cruising operating conditions the propulsion system would realize an energy density of 245 kJ/kg-LN_2 which makes it competitive with the best of lead-acid batteries being used today.

DISCUSSION

Time to recharge (refuel), infrastructure investment, and environmental impact are among the issues to consider, in addition to range and performance, when comparing the relative merits of different ZEV technologies. The convenience of pumping a fluid into the storage tank is very attractive when compared with the typical recharge times associated with lead-acid batteries. Manufacturing LN_2 from ambient air inherently removes small quantities of atmospheric pollutants and the installation of large-scale liquefaction equipment at existing fossil-fuel power stations could make flue gas condensation processes economical and even eliminate the emissions of CO_2.[15] Conversely, liquefaction of air at the local gas station or even at personal residences could provide significant load-leveling to facilitate the efficient generation of electricity. More detailed studies of the consequences of these infrastructure investments would be justified if an effective means of converting the thermal energy potential of LN_2 and liquid air can be developed.

CONCLUSION

The potential for utilizing the available energy of liquid nitrogen for automotive propulsion looks very promising. Heat transfer calculations of a quasi-isothermal reciprocating engine that has a heater core imbedded within its expansion chamber indicate that nearly 85% of the performance of an ideal isothermal power cycle can be attained. A representative case indicates that a 2-cylinder engine operating with an injection pressure of 6 MPa at 850 RPM would produce 15 kW and 190 N-m torque. This power plant would enable a zero emission vehicle to have a range of 140 km from 200 liters of LN_2. Thus even though this expander concept is far from being optimized, it still can provide an effective power plant for a zero emission vehicle.

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C. Knowlen, A. T. Mattick, A. P. Bruckner and A. Hertzberg
University of Washington

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High Efficiency Energy Conversion Systems for Liquid Nitrogen Automobiles

C. Knowlen, A. T. Mattick, A. P. Bruckner and A. Hertzberg
University of Washington

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ABSTRACT

This investigation of the use of cryogens as energy storage media for zero emission vehicles has found that using liquid nitrogen to liquefy the working fluids of one or more closed Rankine power cycles can be an effective means for increasing motive power. System configurations are presented which can realize over 50% of the availability of liquid nitrogen without relying on isothermal expanders. A zero emission vehicle utilizing such a propulsion system would have an energy storage reservoir that can be refilled in a matter of minutes and a range comparable to that of a conventional automobile.

INTRODUCTION

Mitigating increases in urban air pollution and greenhouse gases that come from vehicle tailpipe emissions is the primary motivation behind developing alternative transportation technologies that do not rely on combustion of fossil fuels. Consumer acceptance of a replacement transportation technology, however, is highly dependent on the new vehicle sticker price, operating expenses, reliability, and convenience of use. Thus it is advantageous for the energy storage system of a zero emission vehicle (ZEV) to have low initial cost, be quickly and economically recharged, and to provide driving performance comparable to that of conventional automobiles. In addition, the most desirable technological solutions will eliminate the release of automotive combustion products in areas of poor air quality, while also reducing the net amount of pollutants released to the environment as a consequence of their implementation. Careful consideration of the overall environmental impact of a particular ZEV technology and the corresponding costs of necessary infrastructure development is necessary to evaluate the ability of any new transportation system to meet the goals of society.

Currently, the battery powered electric vehicle is the only commercially available technology that can meet ZEV standards; however, these vehicles have not sold well. This is primarily due to their limited range, slow recharge, and high initial cost. All of these issues can be traced directly to the limitations of electrochemical energy storage, particularly lead-acid and Ni-Cd batteries. These heavy metal energy storage systems remain the dominant technology in the electric vehicle market, but exhibit specific energies in the range of only 30-40 W-hr/kg and energy densities of 60-90 W-hr/l.[1] This compares with about 3,000 W-hr/kg for gasoline combusted in an engine running at 28% thermal efficiency. Lead-acid batteries can take hours to recharge and must be replaced every 2-3 years. Recycling of metals from batteries does significantly reduce the amounts released to the environment; however, these processes are currently not 100% effective. Hence, in addition to significantly increasing operating cost per km driven, the need for several battery replacements over the lifetime of the vehicle raises the specter of increased heavy metal pollution, were lead-acid and Ni-Cd battery powered electric fleets ever to become widespread.[2,3]

The potential of cryogenic energy storage for automotive propulsion has been under investigation at the University of Washington[4,5] and elsewhere[6-9] as an alternative to electrochemical batteries for ZEVs. It is anticipated that the use of an inert cryogen, such as liquid nitrogen (LN_2), for an energy storage medium would not pose any environmental burden, and in particular would avoid the issues of heavy metals pollution associated with Pb-acid and Ni-Cd batteries. Since there exists a well established air manufacturing industry that can be expanded to accommodate large increases in demand, the economy of scale is expected to result in a LN_2 cost that is close to that of the energy consumed in its production (~0.5 kW-hr/kg-LN_2 for a large scale air manufacturing plant). The retail price of LN_2 may be further reduced by credit from additional sales of liquid oxygen and argon that are by-products of the liquefaction process. Straightforward modifications to current gasoline stations would enable LN_2 vehicles to be refueled in minutes and, if the performance and range of cryogen vehicles could be made comparable to gasoline vehicles, the driving habits of motorists would not have to change at all.

Prior work has shown that using LN_2 as the working fluid of an ambient air-heated Rankine power cycle would result in net work output ranging from 50-90 W-hr/kg-LN_2, depending on how isothermal the expansion process can be made.[4,5,10] Even though these specific energy values are superior to those of most electrochemical batteries, the energy density of LN_2 is at best 70 W-hr/l when used in an isothermal expansion process, which results in a 350 liter (90 gal) onboard storage vessel being needed to provide a vehicle with a range comparable to internal combustion (IC) engines. Even though this is not a prohibitively large "fuel" tank, increasing the effective energy density of the LN_2 by using more complex power cycles may be warranted in applications where minimal energy storage volume, zero emission and spark-free operation are required.

In what follows, means to recover work from LN_2 by using it as a heat sink for closed loop Rankine cycles are evaluated. Fundamental thermodynamic considerations are discussed which illustrate the energy storage capabilities of various cryogens. The basic cryogen energy conversion system is presented and applications of topping cycles are examined. The potential gains in specific energy and energy density of LN_2 are determined for a point design that illustrates the implementation of this concept.

THERMODYNAMIC CONSIDERATIONS

The maximum reversible work that can be realized by a chemically inert substance stored at temperature T_l and pressure P_l in an environment at temperature T_a is equal to its availability; $\Psi = h_l - h_a - T_a(S_l - S_a)$. The availabilities of several cryogens of interest and that of compressed air stored at 20 MPa are shown in Table 1. It can be seen that the availability of LN_2 is approximately 3 times that of compressed air and more than 5 times greater than the specific energy of Pb-acid batteries (40 W-hr/kg). The availability of liquid methane (without combustion) is ~50% greater than that of LN_2 on a per mass basis; however, its available energy density is ~25% less than that of LN_2. In the case of compressed air storage at 20 MPa, the available energy density is so low that, even with an ideal isothermal expansion engine, there is not sufficient volume onboard conventional automobiles to provide comparable driving range. Liquefaction of air increases its energy density by several orders of magnitude, resulting in a much more compact energy storage system than that required for compressed air, even when the volume of the accompanying heat exchanger equipment is included.

Table 1: Availability of various working fluids relative to ambient conditions (T_a = 300 K)

Fluid	Storage Temp (K)	Density (kg/m^3)	Ψ (kJ/kg)	Ψ (W-hr/kg)	Ψ (W-hr/l)
N_2	77.4	809	768	213	173
Air	78.9	886	737	205	181
O_2	90.2	1140	635	176	201
CH_4	111.6	423	1093	304	128
C_2H_6	184.6	545	352	97.7	53.2
Compressed Air	300	23.3 233	258	71.5	1.7 17

An energy conversion system which uses ambient heat to vaporize the working fluid inherently reduces the availability of the cryogen in the process. The maximum reversible work that can be generated from any process is equal to the difference between the availability of the working fluid at its peak temperature and pressure and the availability at the end of the expansion process. In the case of an energy conversion system which uses several different working fluids, it is possible to effect a heat exchange process which increases the availability of one medium while decreasing that of another, resulting in a net increase in work output over that of using a single fluid. The availability analysis provides a direct means to evaluate the energy conversion efficiency of various cryogen power system configurations without consideration of many details of the associated processes. Having determined operating conditions of various working fluids which result in the effective use of consumable cryogen (e.g., LN_2, LAir), the mechanical components of an energy conversion system (e.g., heat exchanger area, pressure drop, expansion engine, etc.) can then be designed and the effects of real world inefficiencies evaluated.

CRYOGEN POWER CYCLE

A schematic of a cryogen energy conversion system is shown in Fig. 1. The cryogen "fuel" is stored in a vacuum jacketed vessel which has appropriate relief valves to safely accommodate boiloff. A cryopump pressurizes the fluid to a level somewhat above the injection pressure of the expander to make up pressure loss in the heat exchanger system. Turbines and either rotary or reciprocating fixed-displacement engines are appropriate expanders. The end use of the shaft power typically determines which expander technology is the most appropriate for a given application. Valves having controllable opening times can be used to provide variable cut-off for optimal LN_2 consumption in a piston expander, whereas adjustments in the LN_2 flow rate regulate power output from a turbine. The shaft power from the expander is then readily applied for vehicle propulsion via a standard transmission or the generation of electricity for a hybrid system.

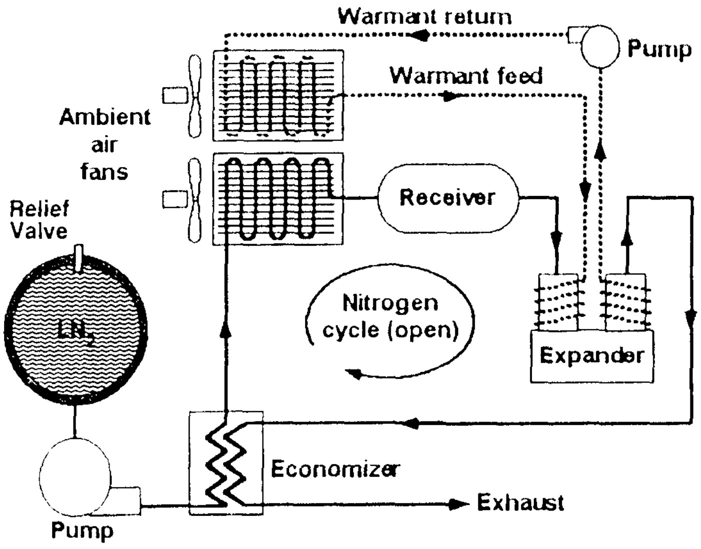

Fig. 1 Schematic of liquid nitrogen energy conversion system.

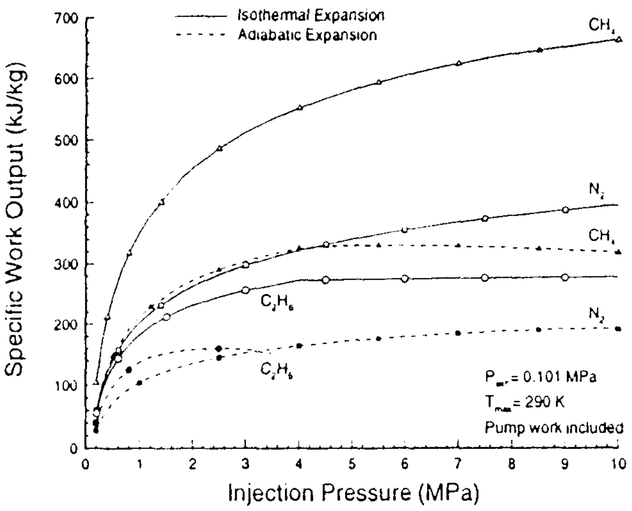

Fig. 2 Specific work of various fluids in adiabatic and isothermal expanders.

Since the rapid expansion process of the cryogen power cycle results in the temperature of the working fluid always being less than ambient, it is possible for a secondary heat transfer fluid ("warmant" in Fig. 1) to conduct sufficient heat into the engine to maintain the expander walls at near room temperature and to effect "quasi-isothermal" operation. Prior investigations have shown that reasonable piston and cylinder head geometries having high surface-to-volume ratios can be used to enhance heat transfer during expansion, resulting in potential net work output of approximately 85% isothermal.[10] An alternative means of approaching isothermal operation is to use multiple expansions with intermediate reheats. Three additional expansions with two reheats result in a net work output of 80-85% isothermal; such triple expansion engines were applied quite successfully in the early part of this century for steamboat propulsion. For the purpose of estimating the maximum energy storage potential of LN_2, the work output from ideal engines operating with adiabatic and isothermal expansion processes can be readily quantified. Since there are several different concepts which can provide quasi-isothermal performance, the work output of a cryogen energy conversion system having a quasi-isothermal expander will be estimated as 85% the isothermal work.

Carefully designed all-weather heat exchangers are required to warm the working fluid to near ambient temperature without suffering a detrimental build-up of frost.[11] An innovative ambient evaporator concept involving multiple coaxial passes of the working fluid has demonstrated the ability to boil and superheat relatively large quantities of LN_2 without any frost accumulation (Williams et al., 1997). The warming of the heat transfer fluid which is circulated around the cylinder heads and walls in a quasi-isothermal expander requires some additional heat exchanger components; however, the increase in power output significantly reduces the rate of LN_2 consumption, which correspondingly reduces the size of the cryogen vaporizer.

CRYOGEN TOPPING CYCLES

The specific energy of several working fluid candidates (N_2, CH_4, C_2H_6) for cryogen power cycles is plotted in Fig. 2 as a function of the injection pressure to both an adiabatic and an isothermal expander. A peak temperature of 290 K is assumed in each case and both the cryopump and expander are assumed to be isentropic. All media are expanded back to atmospheric pressure and the pressure drops in the heat exchanger systems are neglected. The end state of isothermal expansion process is assumed to be 290 K and 0.101 MPa for any injection pressure. The difference in the availability of the gas that is injected into the expander from its availability as it is expelled determines the specific work output.

For all working fluids shown in Fig. 2, the specific work of isothermal expansion from a given injection pressure exceeds that of the adiabatic expander, with the difference being greater as pressure is increased. Relatively little extra work is gained by operating the adiabatic cycles at pressures higher than 3 MPa. The adiabatic work of C_2H_6 is not plotted beyond 3.5 MPa because the end state after adiabatic expansion corresponds to subcooled liquid phase. On a per mass basis it can be seen that the CH_4 cycle produces the most work for a given injection pressure, even though adiabatic expansion to atmospheric pressure drops the temperature to the saturation point when the injection pressure is greater than 4 MPa. The mixed-phase end state might preclude the use of adiabatic reciprocating expanders, but it may not pose an insurmountable problem for turbines. Maxima in the

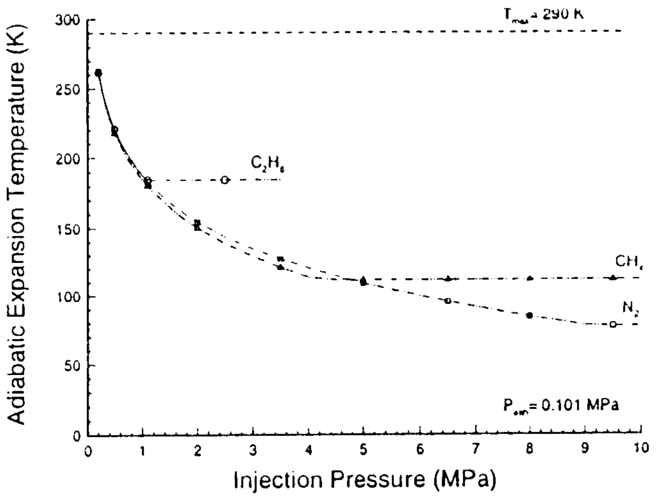

Fig. 3 Temperature after adiabatic expansion to atmospheric pressure.

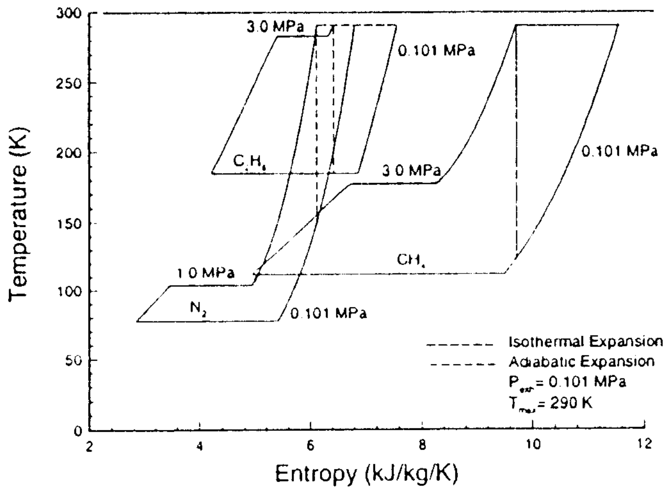

Fig. 4 Cryogen power cycles in the temperature-entropy plane.

adiabatic expansion work for C_2H_6, CH_4, and N_2 are observed at approximately 2, 4, and 9.3 MPa, respectively. The adiabatic work maxima for CH_4 and N_2 occur at the injection pressures which result in the exhaust just reaching the saturation point, as discussed below, whereas the maximum for the C_2H_6 cycle corresponds to a mixed-phase end state.

The adiabatic expansion temperature of these working fluids is plotted versus injection pressure in Fig. 3. Mixed-phase flow after adiabatic expansion, characterized by the horizontal part of the temperature plots, results when the injection pressure is greater than 1, 4, and 9.3 MPa for the C_2H_6, CH_4, and N_2 power cycles, respectively. The enthalpy difference between the end state of the expansion process and that of the saturated liquid at atmospheric pressure is the heat rejection required to completely liquefy the C_2H_6 and CH_4 working fluids. The N_2 is vented back to the atmosphere and reliquefied at a stationary air manufacturing plant. The relatively large differences in saturation temperatures indicate that significant heat transfer between these media is possible and, since the majority of the heat transfer occurs during boiling and condensing processes, the heat exchanger system can be very compact.

The parts of the cryogen power cycles in which the media can exchange heat with a specified minimum temperature differential are readily displayed by temperature-entropy diagrams. An example of an energy conversion system using C_2H_6, CH_4, and N_2 at peak pressures of 3.0, 3.0, and 1.0 MPa, respectively, is shown in Fig. 4. Each of these working fluids is assumed to be stored at atmospheric pressure as saturated liquids. In the case where all of the expanders are adiabatic, the pressurized LN_2 is vaporized and superheated to within 10 K of the exhaust temperature (129 K) of the adiabatically expanded CH_4, then superheated further by heat exchange with the condensing C_2H_6 to approximately 175 K (10 K below C_2H_6 saturation temperature). Ambient heat is then required to bring the N_2 up to the peak temperature of the cycle (290 K). For the calculations presented here, the N_2 exhaust after adiabatic expansion is not used to liquefy any additional C_2H_6 even though there would be some benefit in doing so. The sum of the enthalpy differences between each point of the N_2 heating process along the 1.0 MPa isobar determines the total amount of heat that can be rejected from the other working fluids per kg-LN_2.

The mass flow rate of CH_4 at 3.0 MPa that can be liquefied after adiabatic expansion by vaporizing N_2 at 1.0 MPa is 0.42 kg-CH_4/kg-LN_2. Heat is continuously transferred to the CH_4 at 3.0 MPa as it passes through the C_2H_6 condenser until the CH_4 reaches 175 K. Heat exchange with ambient air then superheats the CH_4 to the peak cycle temperature. Thus the net flow of adiabatically expanded C_2H_6 (from 3.0 MPa) that can be liquefied by the combination of N_2 and CH_4 heat transfer is 0.7 kg-C_2H_6/kg-LN_2.

The heat sink capability of LN_2 for isothermal expanders is determined by assuming 1.0 MPa LN_2 is used to liquefy CH_4 at 290 K and atmospheric pressure. The pressurized CH_4 in turn, liquefies the C_2H_6 flow from a temperature of 290 K. Heat transfer is assumed to terminate in each of the N_2-CH_4 and CH_4-C_2H_6 heat exchangers once the N_2 and CH_4 have been warmed to 280 K. A small amount of ambient heat is then needed to warm the N_2 and CH_4 to the peak cycle temperature of 290 K, whereas the C_2H_6 is completely heated to 290 K with ambient air. Thus the mass flow rate per kilogram of

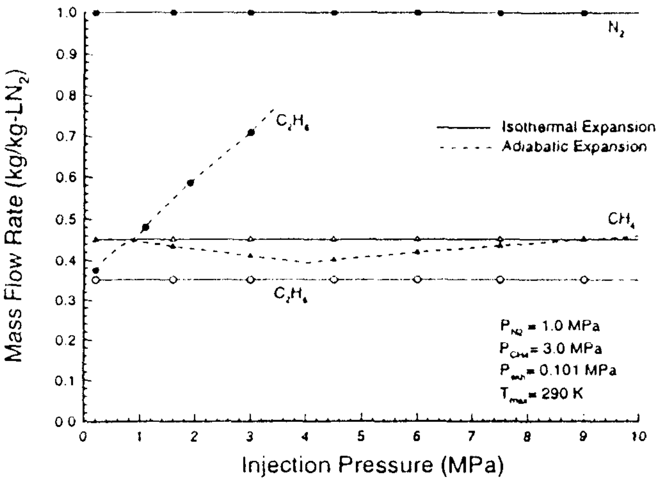

Fig. 5 Mass flow rates relative to 1 kg/s LN_2 at 1 MPa. C_2H_6 flow rate based on that of CH_4 at 3 MPa.

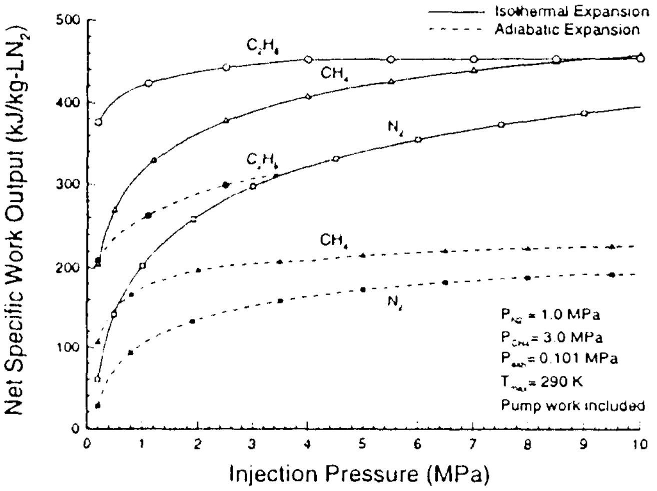

Fig. 6 Net specific work for binary and ternary cryogen energy conversion systems.

LN_2 for a ternary energy conversion system using isothermal expanders under these conditions is 0.36 C_2H_6 : 0.45 CH_4 : 1.0 LN_2.

The mass flow rates of C_2H_6 and CH_4 per kg-LN_2 over a wide range of injection pressures are shown in Fig. 5. Since the end state of the isothermal cycles is independent of the injection pressure, the corresponding flow rates remain constant. The flow rates for the adiabatic expanders are dependent on the heat sink capacities of 1.0 MPa N_2 and 3.0 MPa CH_4. The mass flow rate of CH_4 initially decreases with increasing pressure because the decreasing exhaust temperature after adiabatic expansion reduces the amount of heat that can be transferred to LN_2, even though the amount of heat to be rejected to liquefy the CH_4 is steadily decreasing. Once the end state of the CH_4 expander is at the saturation temperature, the reduction in necessary heat transfer with increasing injection pressure results in the liquefaction of more CH_4 flow. The mass flow rate in the adiabatic C_2H_6 cycle steadily increases with pressure because its saturation point is reached with relatively low expansion and the heat rejection requirement significantly decreases with decreasing quality in the mixed-phase end state.

Fixing the peak pressure for the N_2 allows the net cycle work per kg-LN_2 to be determined for a binary LN_2-CH_4 energy conversion system over a range of CH_4 injection pressures as shown in Fig. 6. Likewise prescribing the N_2 and CH_4 peak pressures enables the effect of pressure on the C_2H_6 topping cycle to be evaluated. The net specific work output of the binary system is the sum of the work outputs from the CH_4 cycles as a function of injection pressure added to the specific work output of the 1.0 MPa N_2 cycle. The addition of a CH_4 topping cycle to the N_2 cryogen power cycle increases the net adiabatic work output by 30-100% in the pressure range of 1-4 Mpa, whereas the isothermal work increases by 25-50% over the same pressure range. At a peak CH_4 injection pressure of 3.0 MPa, the binary system produces 200 and 380 kJ/kg-LN_2 from adiabatic and isothermal expanders, respectively.

The net work output of a ternary energy conversion system using C_2H_6 for the uppermost topping cycle is also shown in Fig. 6 for the case when the peak injection pressures of the LN_2 and CH_4 cycles are 1.0 MPa and 3.0 MPa, respectively. The addition of a C_2H_6 topping cycle boosts the net work output by over 100% in the adiabatic case and by about 80% with an isothermal expander. Thus it is possible to extract over 300 kJ/kg-LN_2 using adiabatic expanders and over 450 kJ/kg-LN_2 with isothermal expanders. This latter value indicates that a quasi-isothermal expander may achieve net specific work of over 400 kJ-kg-LN_2 (110 W-hr/kg-LN_2 or 90 W-hr/l-LN_2), which enables an availability utilization of over 50% while keeping the peak system pressure at 3.0 MPa.

In order to illustrate how to best use the LN_2, the effect of varying the N_2 injection pressure on the net specific work of a system having a peak pressure of 3.0 MPa for both the C_2H_6 and CH_4 topping cycles is shown in Fig. 7. Adiabatic and isothermal work of just the N_2 power cycle are also plotted. The work output for the binary and ternary systems when no work is generated by the vaporized N_2 (P_{inj} = 0.101 MPa) is indicated by the left most plotting symbols. Not enough cases are plotted for these topping cycle parameters to resolve the performance of this system for N_2 injection pressures below 1.0 MPa; however, it is apparent from the negative slope of the topping cycle work curves that LN_2 is best used at lower pressure where it produces less work by itself yet has more heat sink capability. The best use of CH_4 in an isothermal system

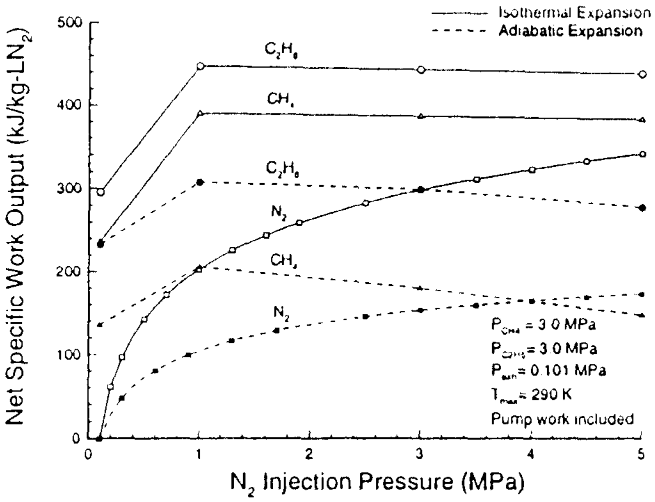

Fig. 7 Net specific work for binary and ternary systems when injection pressure of topping cycles is fixed at 3.0 MPa.

was also found to be when it is used at injection pressures lower than 3.0 MPa (~500 kJ/kg-LN$_2$); however, the net work output of the adiabatic ternary system was not significantly affected. Thus the performance of the configuration presented in Fig. 6 is a reasonable estimation of what is possible for a ternary cryogen energy conversion system using these three media.

DISCUSSION

A significant operational aspect of a multi-fluid cryogen energy conversion system is that the minimum temperatures of the fluids within the ambient heat exchanger are at the condensation point of the uppermost topping cycle fluid. This increases the total surface area required to transfer the necessary amount of ambient heat into the system, yet minimizes the degree of difficulty of preventing frost accumulation because of the corresponding increase in the minimum temperature of the flows within the ambient heat exchanger. Hence, in humid environments where large flow rates of LN$_2$ are required to be vaporized, the additional expense and complexity of adding topping cycles may be warranted to enable long periods of frost-free operation, in addition to increasing the LN$_2$ utilization efficiency.

The choice of CH_4 and C_2H_6 for use in this work was primarily due to their saturation properties at atmospheric pressure. It is recognized that there are many organic compounds being applied in refrigeration processes that have favorable properties for cryogen energy conversion systems, and thus there most likely exist more effective combinations of working fluids and operating conditions that can better utilize LN$_2$.

There are several ways to couple the shaft power from multiple expanders to an automotive drive train. Such systems are currently in use for many kinds of hybrid electric vehicles. Conversely, the expanders can all be connected to a common shaft, whether they are rotary or reciprocating devices. For example, a six cylinder engine may be configured to expand three different working fluids through three pairs of adjacent cylinders to minimize imbalance due to variation in mean effective pressures. Loss of working fluids in the closed loop cycles could probably be made up on a maintenance schedule similar to that for changing oil in conventional vehicles. The relatively low peak pressure and temperature of the system reduces the expander wall thickness requirements and thus the thermal impedance and engine mass are also reduced. Transient response times and system start up scenarios need to be investigated in more detail to better quantify the operating characteristics of these cryogen energy conversion systems. It is apparent that there are several complex issues yet to be resolved before an automotive cryogen power plant using multiple fluids can be implemented; however, there do not seem to be any fundamental inhibiting factors that preclude the eventual development of such systems.

CONCLUSIONS

The use of liquid nitrogen as a heat sink for several cascaded topping cycles has been examined. Operating conditions for binary (LN$_2$-CH_4) and ternary (LN$_2$-CH_4-C_2H_6) systems have been identified which can realize specific energies in the range of 200-380 kJ/kg-LN$_2$ and 300-450 kJ/kg-LN$_2$, respectively, depending on the degree of isothermal expansion that can be realized. This results in the LN$_2$ being used with nearly twice the efficiency of a single expander operating at the same injection pressure. In general, it was found that the best use of LN$_2$ and CH_4 was at relatively low injection pressures (< 1.0 MPa) in order to maximize utilization of their heat sink capabilities.

ACKNOWLEDGMENTS

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C. Knowlen, A. T. Mattick, A. Hertzberg and A. P. Bruckner
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Ultra-Low Emission Liquid Nitrogen Automobile

C. Knowlen, A. T. Mattick, A. Hertzberg and A. P. Bruckner
University of Washington

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ABSTRACT

Means to extend the range of cryogen (liquid nitrogen or liquid air) powered automobiles via burning a small amount of fossil fuel (gasoline or liquid methane) have been investigated. By utilizing both an ambient air-heat exchanger to vaporize the cryogen and a fossil fuel-fired superheater to elevate the temperature of the gaseous product, the range of the vehicle can be three times that of an ambient-heated propulsion system while not exceeding current ultra-low emission standards. Internal and external combustion power cycles using either liquid air or nitrogen as the working fluid were found to be more fuel efficient than an internal combustion engine operating on the standard Otto cycle. The fuel-cryogen operating expense for the proposed hybrid propulsion systems was found to be higher than that of the conventional automobile; however, the performance calculations were very conservative. Fossil fuel consumption per mile of dual-fuel systems are projected to be over 40% less than that of conventional gasoline automobiles, thus offering the potential of significant reductions in greenhouse gas and pollutant tail pipe emissions.

INTRODUCTION

It is now apparent that zero emission vehicle (ZEV) technologies relying on electro-chemical batteries for energy storage are a long way from providing the driving performance and low sticker price desired by the majority of the populace, as evident by the lease-only marketing strategy for the EV1 in the state of California.[1] Consequently, the impetus in technology development has shifted from ZEV to ultra-low emission vehicles (ULEV), which strive to provide reasonable driving performance from either hybrid combustion/electric or fuel cell propulsion systems while releasing minimal combustion products.[2,3]

The potential of using cryogenically stored gases in external and internal combustion power cycles of ULEVs is currently under investigation at the University of Washington. Storing the working fluid of a power cycle as a liquid offers a significant advantage in producing high working pressure because of the relatively low energy needed for pressurization. Earlier work at the University of Washington[4] and the University of North Texas[5,6] has shown that liquid nitrogen can be a very effective working fluid for zero emission vehicles employing only ambient air as the source of heat. Issues related to ambient heat exchange under all weather conditions,[7] quasi-isothermal expansion,[8] and the use of ternary fluid systems for maximum round trip energy efficiency[9] have been explored. These studies indicated that LN_2 powered automobiles can compete favorably with electro-chemical energy storage systems on the basis of operating costs, range, user friendliness, readiness of key technologies, and environmental benefits. In recognition of the need for ULEV performance comparable with that of current automobiles, the potential of augmenting the power output of cryogenic propulsive systems with the addition of combustion heat release motivated the present study.

Prior work by Latter *et al.*[10] had shown that significant improvements in fuel economy and reduced emissions were feasible using a fossil fuel-fired vaporizer to heat pressurized liquid nitrogen to well above ambient temperatures, and to provide subsequent reheats in a triple expansion motor. They also proposed combining fuel with pressurized air to effect constant volume combustion in a piston expander to enhance vehicle range.[11] The mileage of such vehicles was estimated to be about 90 mpg-diesel and 3 mpg-liquid air. Although the performance predictions were encouraging, no vehicle prototypes using these power systems are known to have been constructed.

Analyses of the potential performance of various propulsive cycles have been carried out to ascertain the viability of using onboard cryogenically stored gases as the primary working fluid of automotive propulsion systems. In particular, Otto and Rankine cycle scenarios have been examined for both reacting and inert primary working fluids. An "air-standard" engine model, with an assumed mechanical efficiency, was employed to evaluate the work output of different cycles for comparison purposes. Details of the thermodynamic cycles under consideration and the numerical modeling procedure are provided in next section. The contribution to the net specific work of each process of the propulsive cycles are examined in the results section. The projected fuel consumption rates and operating costs of these cycles are presented for air and N_2 working fluids using gasoline and methane has potential fuels.

THERMODYNAMIC CYCLE MODELING

A relatively simple model of an internal combustion engine based on an Otto cycle which incorporates mechanical inefficiencies is developed. This model is applied to dual-fuel concepts which have constant volume and constant pressure heat addition processes to superheat a vaporized cryogen that is used as the working fluid of the power cycle. The mechanical efficiencies of all expansion processes in each of the power cycles are assumed to be the same, as is the efficiency of all compression processes.[12] A combustion efficiency parameter is incorporated to account for the phenomena which reduce the utilization of the full heating value of the fuel in both internal combustion engines and externally-fired heaters. Details of the modeling procedure are provided below.

STANDARD OTTO POWER CYCLE – Automobile engines typically operate with a four-stroke power cycle involving steps for fuel-air intake, compression, power, and exhaust gas expulsion. These processes for an Otto cycle can be represented in the pressure-volume plane as shown in Fig. 1. Engines operating with this power cycle have been highly developed over the years. There are several fundamental drawbacks to the Otto cycle, however, which limit its thermal efficiency. In the first place, the back work required to compress the fuel-air charge (path 1-2) reduces net work output. In addition there is parasitic work required to expel the exhaust gases (path 5-6) and to draw in the new air charge (path 7-1). Finally, since the engine operates with a fixed compression ratio (V_1/V_2), the working fluid cannot be fully expanded to maximize the conversion of its availability to useful work. Unavoidable heat transfer losses are also significant in reducing the net work output; however, this concern is typically shared by many heat engines.

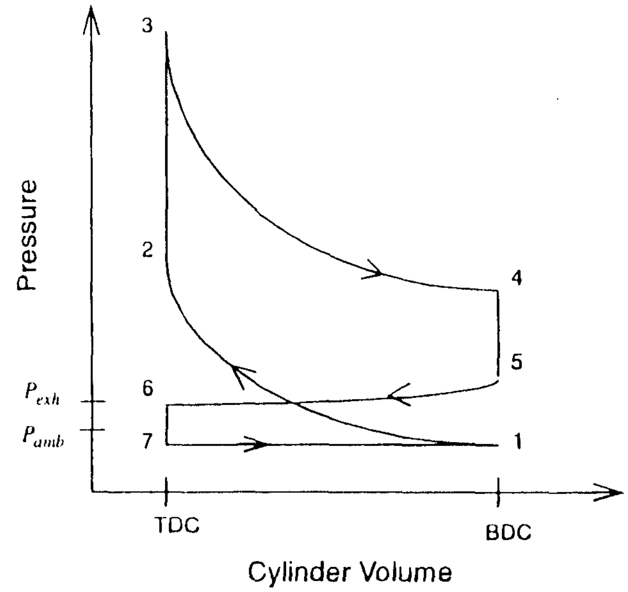

Fig. 1 Standard four-stroke Otto power cycle.

The net work output of the standard Otto cycle is determined by summing the work done on and by the piston for each part of the power cycle (Fig. 1). This allows different mechanical efficiencies to be assigned to each process in order to account for pertinent real world effects. The mechanical efficiency for the expansion stroke, η_e, is defined as the ratio of actual work out for a given pressure change, w_{exp}, to the ideal work of an isentropic expansion through the same pressure change, w_{34s}. Thus the work done on the piston by the gas during the expansion process between states 3 and 4 (Fig. 1) is:

$$w_{exp} = \eta_e\, w_{34s} = \eta_e\, c_v\, T_3 \left(1 - (P_4/P_3)^{(\gamma-1)/\gamma}\right) \qquad (1)$$

where c_v is constant volume specific heat capacity, T_3 is peak cycle temperature, P_3 is peak cycle pressure, and γ is the specific heat ratio. Conversely, the compression mechanical efficiency, η_c, is defined as the ratio of the isentropic compression work input for a given pressure change, w_{12s}, to the actual work input, w_{comp}, as follows:

$$w_{comp} = w_{12s} / \eta_c = c_v\, T_1 / \eta_c \left((P_2/P_1)^{(\gamma-1)/\gamma} - 1\right) \qquad (2)$$

The total work done on the air in the crankcase by the piston during the expansion and compression strokes is assumed to be zero in this part of the analysis.

After the power stroke, the gas is assumed to immediately vent down to the exhaust pressure of the system, $P_{exh} = P_5$ (i.e., constant volume heat rejection). For the purposes of this study it will be assumed that the mechanical efficiencies of the intake and exhaust strokes are the same as those for expansion and compression, respectively. The intake and exhaust strokes are considered as isobaric volume changes whose actual work values are given as:

$$w_{int} = \eta_e\, (P_{int} - P_{amb})\, (V_1 - V_2) \qquad (3)$$

$$w_{exh} = (1/\eta_c)\, (P_{exh} - P_{amb})\, (V_1 - V_2) \qquad (4)$$

where P_{int} is the intake manifold pressure, P_1, and the crankcase pressure is equal to P_{amb}. The sign convention adopted here is for work out to be positive and work in to be negative. The sum of the work of the four strokes equals the net work output of the engine per unit mass of working fluid as follows:

$$w_{net} = w_{exp} - w_{comp} + w_{int} - w_{exh} \qquad (5)$$

The heat input during the combustion process, q_{23}, is determined from:

$$q_{23} = c_v\, (T_3 - T_2) \qquad (6)$$

where: $T_2 = T_1 \left(1 + 1/\eta_c \left((P_2/P_1)^{(\gamma-1)/\gamma} - 1\right)\right)$

The amount of fuel that needs to be added for a given amount of heat input is determined from the lower heating value of the fuel (H_{lhv}) and a combustion efficiency, η_b, which accounts for all non-idealities of the fuel burning process and the heat lost to engine cooling. Thus the fuel mass flow rate, m_f, required to provide the heat input per unit mass of working fluid is given as:

$$m_f = q_{23} / (\eta_b H_{lhv}) \qquad (7)$$

One other mechanical efficiency that could be included is that of the transmission drive train of the vehicle. This value will be neglected, however, because it is assumed that the crankshafts of all the engine concepts considered here are coupled to the same drive train. By adjusting the mechanical efficiencies of each step outlined above, this model of an Otto cycle powerplant can be tuned to predict the fuel economy of a representative vehicle. The corresponding efficiency values are then used for mechanical components and heat transfer equipment of alternative power cycles being considered.

MODIFIED OTTO POWER CYCLE – Once the possibility of storing liquid air on board a vehicle is introduced, then an intriguing variation of the Otto power cycle can be considered. The schematic of a propulsion system using liquid air as the working fluid for a modified Otto power cycle is shown in Fig. 2. Key components are: liquid air storage dewar, cryogen pump, ambient air heat exchanger, engine exhaust recuperator, reciprocating expander, and hydrocarbon fuel handling system.

The pressure-volume diagram of this propulsion system is shown in Fig. 3 with two horizontal axis indicating the pressure as a function of both specific volume, v, and cylinder volume, V. Operation begins by first pumping liquid air from storage (state 0) to a pressure (state 1) somewhat higher than that resulting from the typical compression stroke of the Otto cycle to compensate for pressure drop in the ambient air heat exchanger. The pressure drop through the ambient heat exchanger, ΔP, is assumed to be proportional to the pump outlet pressure, i.e., $\eta_p = \Delta P / P_1$, where η_p is the pressure drop coefficient. Ambient heat exchange vaporizes and superheats the liquid air to near ambient temperature, after which further superheat is possible from the relatively warm engine exhaust. The resultant air is injected with pressurized fuel into the clearance volume, V_{clr}, of the cylinder as the piston approaches TDC at the same P_2 of the Otto cycle (Fig. 1). The injection of fuel-air mixture continues until a sufficient amount of working fluid is admitted (path 2'-2). Spark ignition of the fuel-air mixture is then initiated and heat release occurs under constant volume conditions (path 2-3) to generate peak cycle pressure. With the appropriate choice of expansion ratio, V_4/V_3, the combustion products can be fully expanded to the back pressure of the exhaust system, P_{exh}.

The net work output of this cycle is determined by summing the work of each process in the same manner as before. The work of the isobaric fuel-air injection process, W_{inj}, is assumed to be transmitted to the crankshaft with the same mechanical efficiency as the expansion stroke in the standard Otto cycle, and is determined by:

$$W_{inj} = \eta_e (P_2 - P_{amb}) (V_2 - V_{clr}) \qquad (8)$$

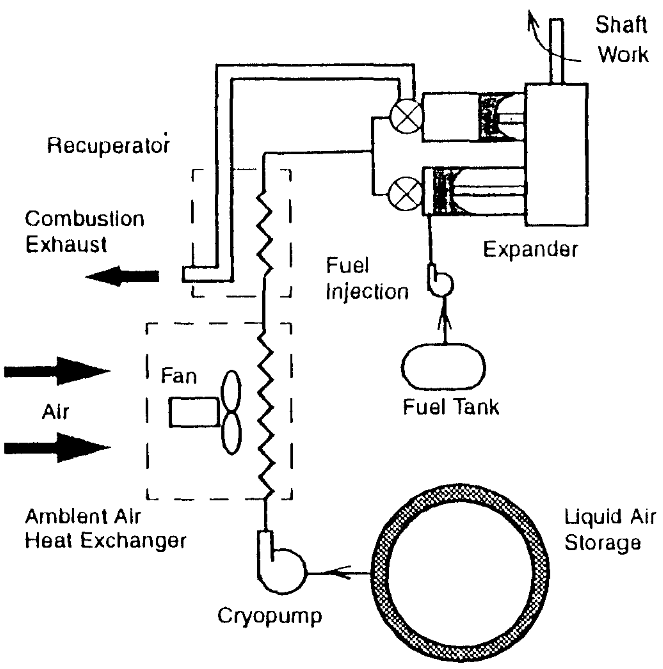

Fig. 2 **Modified Otto cycle propulsion system using liquid air with internal combustion.**

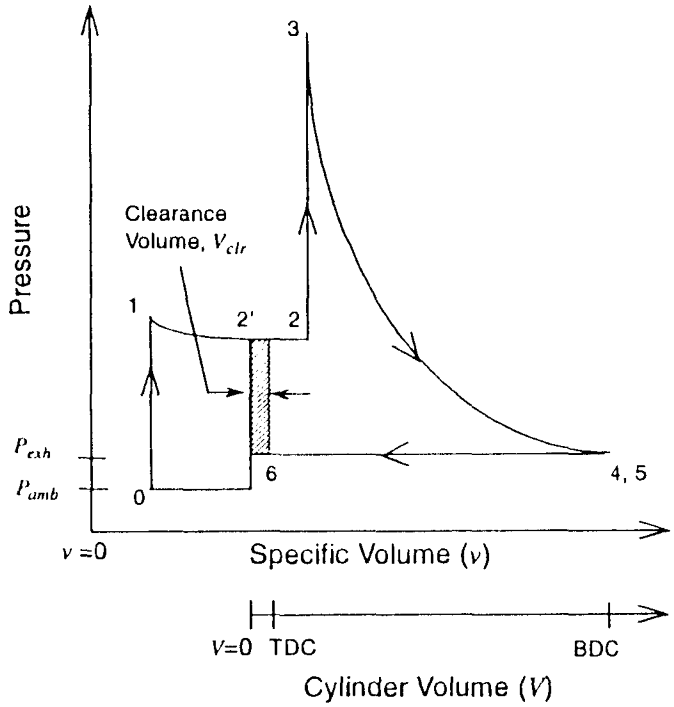

Fig. 3 **Modified Otto power cycle with internal combustion.**

where again the crankcase pressure is assumed to be equal to ambient. Exhaust work is similar to that of the standard Otto cycle; however, the clearance volume must be accounted for as follows:

$$w_{exh} = (1/\eta_c) (P_{exh} - P_{amb}) (V_4 - V_{clr}) \qquad (9)$$

The cryogen pump work, w_{pump}, is determined for a constant volume process with pumping efficiency, η_p, as shown here:

$$w_{pump} = v_0 (P_1 - P_0) / \eta_p \qquad (10)$$

where v_0 is the specific volume of the cryogen in the storage dewar.

Since there is no intake work (the filling of the clearance volume does not require work), this propulsion system is effectively a two-stroke power cycle. The work of the piston on the air in the crankcase is now included in the expansion stroke as follows:

$$w_{exp'} = w_{exp} - P_{amb} (V_4 - V_3) \qquad (11)$$

Thus the net specific work is determined:

$$w_{net} = w_{exp'} + w_{inj} - w_{pump} - w_{exh} \qquad (12)$$

The heat input to the cycle can be determined from Eq. 6 when no recuperator is used. For purposes of comparing the energetics of the power cycles considered here, the potential benefits of utilizing exhaust heat in the propulsion system will be ignored. Thus the fuel mass flow rate is given by Eq. 7.

Increased work output potential exists for the internal combustion Rankine cycle over the standard Otto cycle for several reasons besides the use of liquid pumping versus gas compression. If the peak temperature is specified and the incoming air is at near room temperature, the increase in temperature difference from a constant volume combustion process produces higher peak pressure. Higher effective compression ratio may also be feasible because the lower pre-ignition temperature of the fuel-air mixture is less prone to detonate, thus reducing the need for higher octane gasoline. Another significant factor is that the expansion ratio is not constrained by the compression process, so that near optimal expansion is possible.

EXTERNAL COMBUSTION MODIFIED OTTO CYCLE – The schematic of a propulsion system which uses external combustion for constant volume heat addition and an inert working fluid (N_2 or air) is shown in Fig. 4. Maximum system pressure is generated in a firebox which pressurizes individual tubing coils while the isolation valves are closed. Multiple pressure-building coils for each engine cylinder could be used as a means to overcome the longer heating duration required to warm gases in a tube versus that of an internal combustion process. The possibility of preheating the ambient temperature working fluid with combustor and engine exhausts is indicated on Fig. 4.

The pressure-volume diagram of this constant volume, external combustion power cycle is shown in Fig. 5. The vaporization of the cryogen working fluid is accomplished in the same manner as described for the previous power cycle. Pressure drop in the ambient heat exchanger is also accounted for as previously described. After being vaporized and superheated to near ambient temperature (state 2'), the N_2 gas is admitted to individual pressure-building coils having relatively small volume, V_{pbc}, as indicated in Fig. 5. Constant volume heat addition rapidly increases the internal energy of the gas between states 2 and 3 until the cylinder intake valve is opened. The gas fills the clearance volume and then expands against the receding piston until the exhaust pressure is reached at state 4. After the piston reaches BDC the exhaust valve is opened and the gas is pushed out of the cylinder and through the exhaust system at pressure $P_4 = P_{exh}$.

The work output and input of the expansion and exhaust strokes, respectively, are determined with Eqs. 1, 4 and 11, assuming the same mechanical efficiencies as before. The work input from the cryogen pump is determined from Eq. 10. Even though the heat input to the pressure-building coil tends to reduce the rate of pressure drop during the expansion stroke, the expansion process is assumed to be adiabatic. The volume expansion ratio, $V_4/(V_{clr} + V_{pbc})$, is chosen to fully expand the peak cycle pressure. No injection work is available in this cycle, thus the net work is determined from Eq. 12 with $w_{inj} = 0$. Since no heat exchange with the exhaust flows is considered, the heat input and fuel mass flow rate are determined from Eqs. 6 and 7, respectively. Although such a constant volume heater is unusual for propulsive power, this approach does serve as a convenient means to compare potential energetic advantages of internal versus external heat addition processes for cryogenic working fluids.

CRYOGENIC RANKINE POWER CYCLE – The maximum cycle pressure of a Rankine power cycle using a liquefied air product for the working fluid is usually generated with a cryogen pump (pressure-building circuits are an alternative). The heat addition process is ideally isobaric and the expansion process can be adiabatic or isothermal (ideally), depending on peak cycle temperature. The propulsion system schematic in Fig. 4 applies to an open Rankine cycle, if the isolation valve between the ambient heat exchanger and the firebox is eliminated. Another version of a cryogenic Rankine power system is shown in Fig. 6 which illustrates a double expansion with single reheat, along with several heat exchange processes involving the exhausted working fluid. Again the exhaust heat exchange and reheat processes are illustrated for generalization purposes, but the analysis is conducted for only the basic cycle without reheat or recuperation.

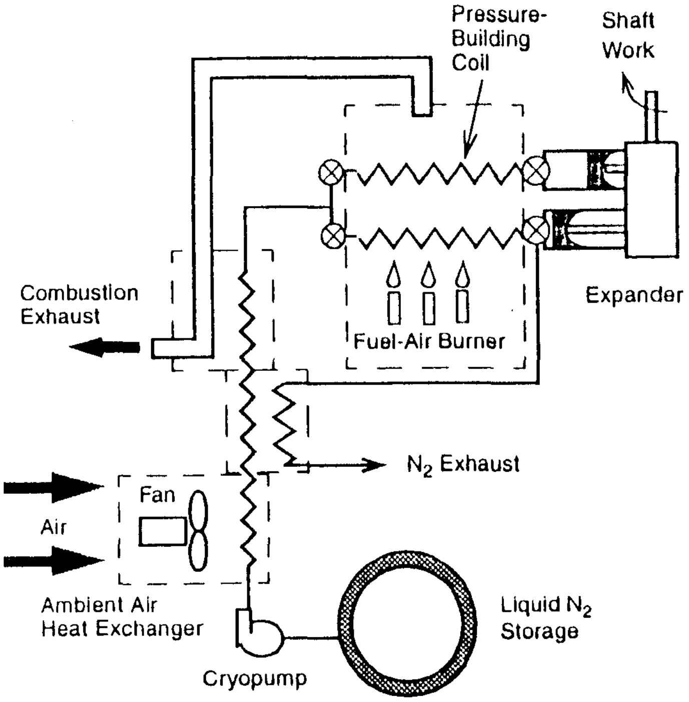

Fig. 4 **Modified Otto cycle propulsion system with external heat addition.**

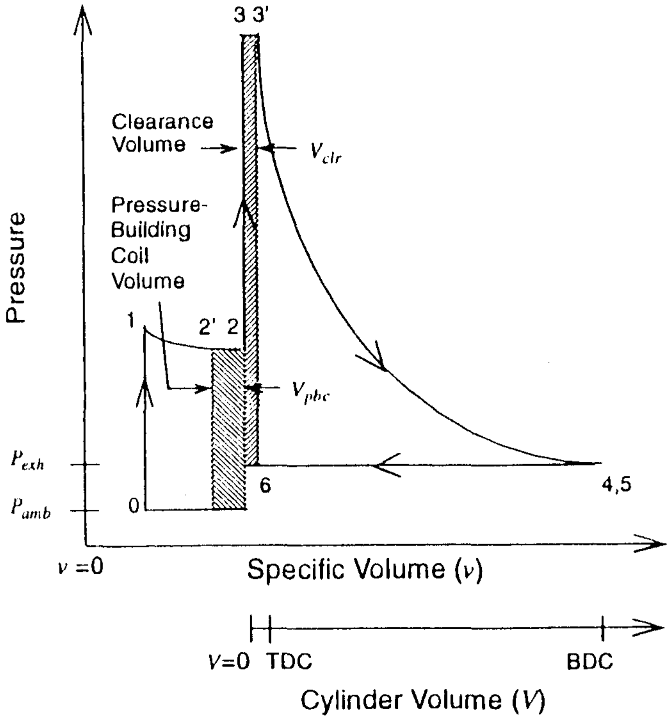

Fig. 5 **Modified Otto power cycle with external combustion.**

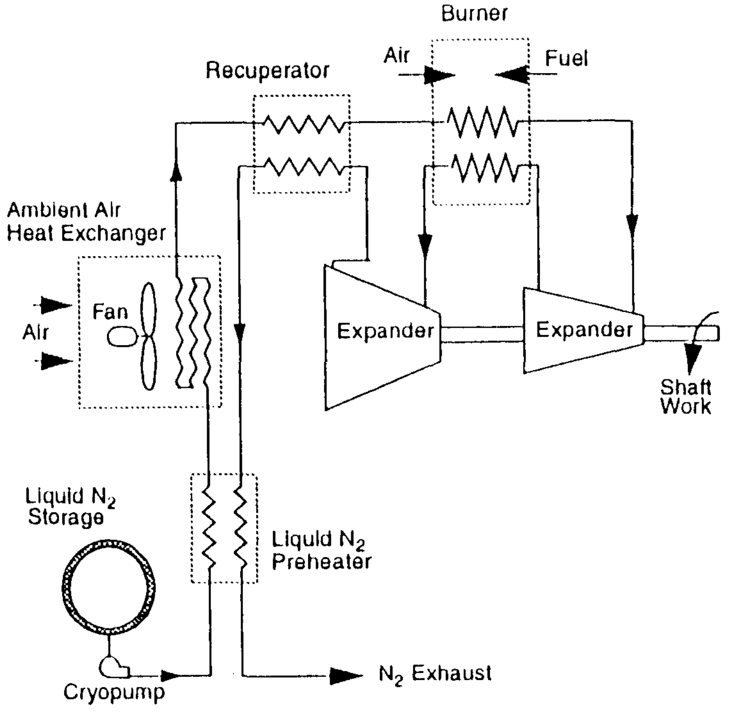

Fig. 6 **Cryogenic Rankine propulsion system with external heat addition.**

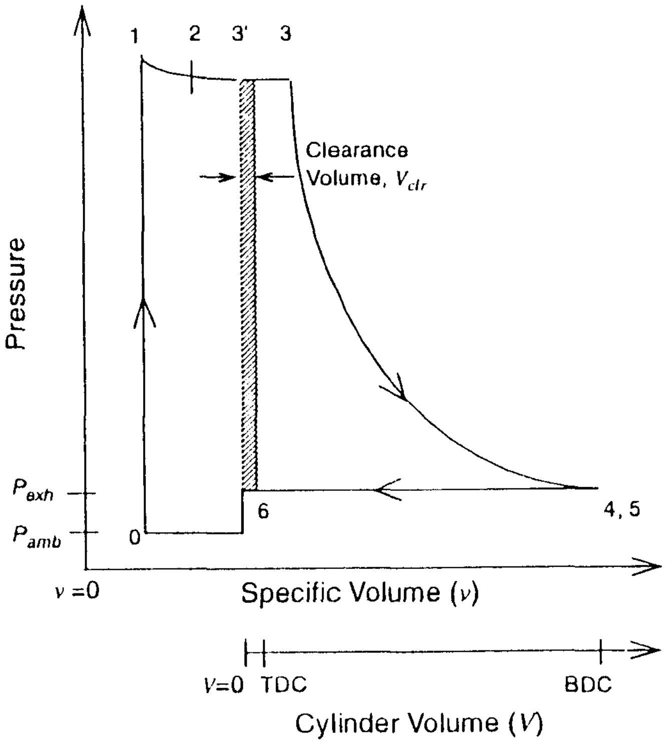

Fig. 7 **Cryogenic Rankine cycle with external heat addition.**

The pressure-volume history of the cryogenic Rankine power cycle without reheat is shown in Fig. 7. The process for cryogen pressurization via pumping and vaporization in the ambient heat exchanger is as previously described. Significant superheat is provided at nearly constant pressure in the firebox along the path 2-3'. The gas is then injected into the expander and immediately fills the clearance volume as the piston approaches TDC. Injection at constant pressure continues until the piston has receded to the point of optimum cutoff, $(V_3 - V_{3'})/V_4$, producing injection work determined from Eq. 8. The work of the adiabatic expansion process and exhaust stroke are determined as previously described. Thus the net work of the cryogenic Rankine power cycle is determined from the following:

$$w_{net} = w_{exp'} + w_{inj} - w_{pmp} - w_{exh} \qquad (13)$$

Fuel consumption for this power cycle is based on the lower heating value of the fuel H_{lhv} and combustion efficiency η_b as before. However, the total heat input is now for a constant pressure process, i.e.,

$$q_{23'} = c_p (T_{3'} - T_2) \qquad (14)$$

where c_p is the constant pressure specific heat capacity. The heat transfer efficiency of the burner is assumed to be the same as the combustion efficiency in the Otto cycle. Thus the fuel mass flow rate is determined from Eq. 7 while using the heat input from Eq. 14.

It should also be noted that in the absence of external-combustion heat addition this power cycle corresponds to the ZEV mode of operation. Under these circumstances, the cutoff ratio is adjusted for optimum expansion and mechanical efficiencies are assumed to remain constant. The ability to easily switch to ZEV operation by shutting down the fuel flow and adjusting the cutoff is an attractive feature of the cryogenic Rankine cycle just described as well as with the internal and external combustion modified Otto cycles.

THEORETICAL RESULTS

The net specific work output of the four fuel-fired engine cycles described in the previous section are determined as functions of volume compression or expansion ratio, V_1/V_2 or V_4/V_3, and peak cycle pressure, P_3, with peak temperature, T_3, as an input parameter. The mechanical efficiency of the individual process steps is assumed to be the same for the corresponding process of each power cycle. For the numerical results presented here, the mechanical efficiencies of the expansion and compression processes are assumed to be equal and to have a value of $\eta_e = \eta_c = 0.85$, whereas the mechanical efficiency of the cryogen pump is assumed as $\eta_p = 0.50$. The ambient heat exchangers are assumed to vaporize and superheat the cryogen to within 95% of the ambient temperature with a 10% pressure drop due to heating and viscous losses ($\eta_p = 0.10$). The combustion efficiency of the internal and external heat addition processes is $\eta_b = 0.70$. A clearance volume of $V_{clr} = 0.02$ is used in all calculations. These values are chosen because they are representative of currently available technology and provide a self-consistent means of evaluating the relative performance merits of the power cycles.

The primary working fluid, N_2 or air, and fuel, methane (CH_4) or octane (C_8H_{18}), are stored at atmospheric pressure, $P_0 = P_{amb}$, as liquids. Physical properties for the fluids of interest are shown in Table 1.[13] The specific heats are given for the gaseous phase at a temperature of 300 K. The lower heating value of the fuel is used to provide conservative estimates of fuel consumption. The fuel-air mass ratios are assumed to be so small that the working fluid properties are primarily those of the vaporized cryogen. Since the primary purpose of this study is to compare different power cycles, temperature effects on heat capacity are neglected and the specific heat ratio at 300 K is used for all calculations.

Table 1. Properties of various cryogens and fuels

Fluid	Storage T_o (K)	Density (kg/m^3)	c_p (kJ/kg/K)	c_v (kJ/kg/K)	H_{lhv} (MJ/kg)
N_2	77.4	809	1.042	0.745	0
Air	78.9	886	1.004	0.717	0
CH_4	111.6	423	2.254	1.735	50.01
C_8H_{18}	300	706	1.711	1.639	44.43

INTERNAL COMBUSTION CYCLE PERFORMANCE – The specific work output for each individual stroke of the standard Otto cycle model is plotted in Fig. 8 as a function of volume compression ratio for a peak cycle temperature of $T_3 = 2000$ K. The working fluid in this case is ambient air and the fuel is pure octane. The intake (P_1) and exhaust (P_{exh}) pressures are 0.091 MPa and 0.11 MPa, respectively. The intake (w_{int}) and exhaust (w_{exh}) works (right vertical axis) are seen to be relatively insensitive to compression ratio when V_1/V_2 exceeds a value of 5. Note that work input is required to draw in the fuel-air mixture for these operating conditions. The compression work (w_{comp}) increases at a faster rate than the expansion work (w_{exp}), resulting in decreasing net specific work (w_{net}) at compression ratios greater than 7. This indicates that the most efficient use of air would be realized with a compression ratio of $V_1/V_2 \sim 7$; however, since air is "free" it does not necessarily follow that this is the optimum compression ratio for minimum operating expense.

The specific fuel consumption per unit of net work output for the standard Otto cycle is shown in Fig. 9 for four different peak temperatures. It can be seen that the fuel consumption decreases as the peak temperature increases. Higher peak temperature shifts the minima of the specific fuel consumption curves to higher compression ratio and reduces their sensitivity to increasing V_1/V_2. The flame temperature of octane-air combustion is typically around 2200 K,[14] thus the bounding curves with $T_3 = 2000$ and 2500 K indicate that engines with compression ratios of 10-15 would be the most fuel efficient, which is consistent with current practice. It is of interest to note that even if pre-ignition could be avoided under all operating conditions, there is not much more fuel efficiency to gain by increasing the compression ratio of current spark-ignited engines beyond 12 or so.

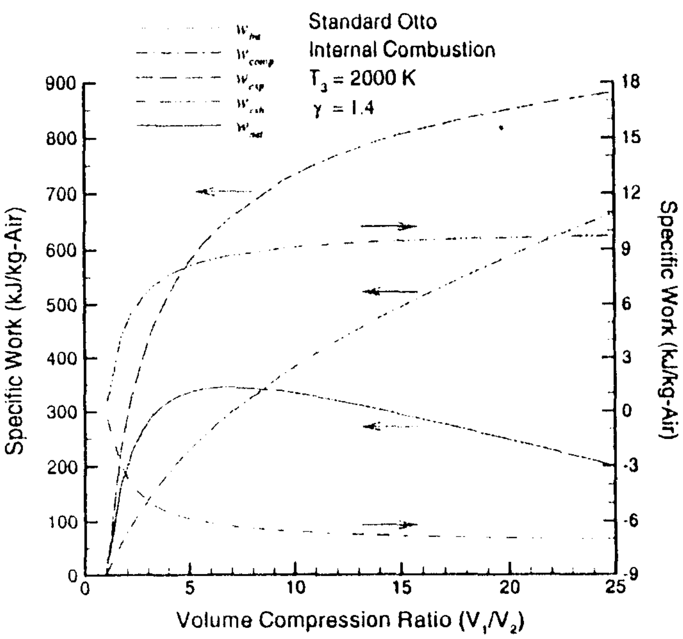

Fig. 8 Specific work of standard Otto cycle.

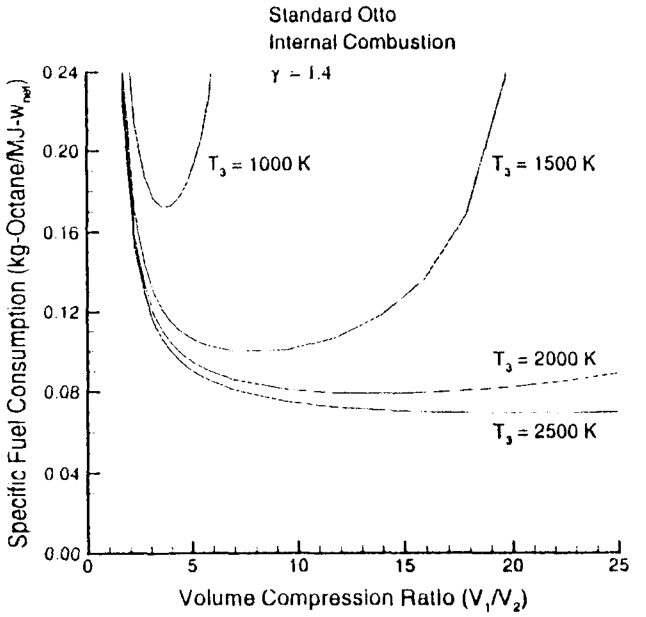

Fig. 9 Specific fuel consumption of standard Otto cycle.

The specific work for each process of the modified Otto cycle as a function of the volume expansion ratio (V_4/V_3), with internal combustion using liquid air (LAir) as the working fluid, is shown in Fig. 10. The peak temperature is T_3 = 2000 K and the peak pressure (P_3) increases with increasing expansion ratio. The cryopump work (w_{pump}) monotonically increases with expansion ratio although its effect on the net work is negligible over the range shown. Even though the exhaust pressure remains constant, the exhaust work decreases because the specific volume of the air at the end of the expansion stroke (v_4) decreases with increasing peak pressure. Thus the actual cylinder volume (V_4) needed to process a unit mass of working fluid decreases with increasing V_4/V_3 when the peak temperature is kept constant. The injection work (w_{inj}) has a maximum because at higher expansion ratio the injection process is cut off at smaller V_2 and a larger fraction of the gas is used to fill the clearance volume (V_{clr}). The work done on the air in the crankcase (not shown) negates the injection work and thus the net work is less than that of the expansion stroke throughout the range of expansion ratio displayed here. The net work per unit mass of air monotonically increases and is about twice that of the standard Otto cycle at expansion ratios greater than 10. Thus this engine would be smaller than a conventional one having the same power output and revolution rate.

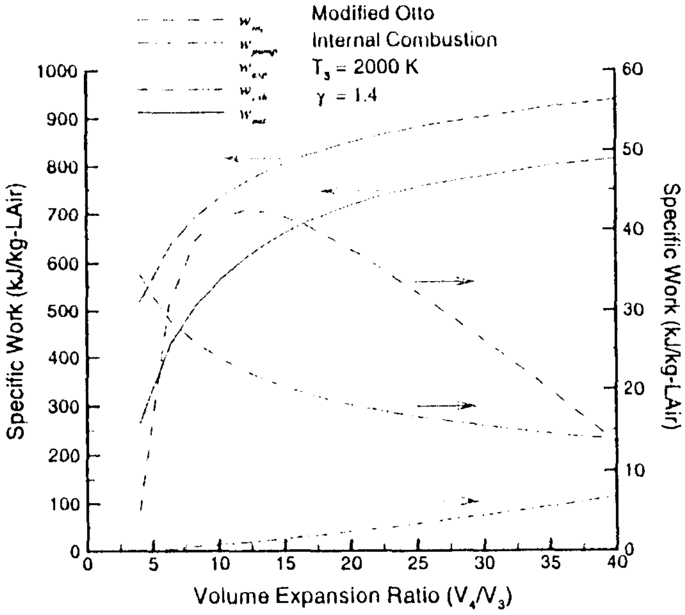

Fig. 10 Specific work of modified Otto cycle with internal combustion using liquid air.

The effects of peak temperature on the fuel economy of the modified Otto cycle with internal combustion are indicated by the specific fuel consumption curves in Fig. 11. Contrary to the standard Otto cycle, the fuel economy improves with decreasing peak temperature because less fuel is needed to heat the working fluid. Consequently the engine can be designed to operate cooler which reduces its heat rejection requirements and the level of nitrogen oxide emissions from the combustion process. The fuel consumption of the internal combustion modified Otto cycle at a representative operating point of T_3 = 2000 K and V_4/V_3 = 20 is about 33% less than that of the standard Otto cycle engine having typical operating conditions of T_3 = 2000 K and V_1/V_2 = 10. Thus the modified Otto engine has the potential to be more compact and fuel efficient while emitting fewer pollutants during operation.

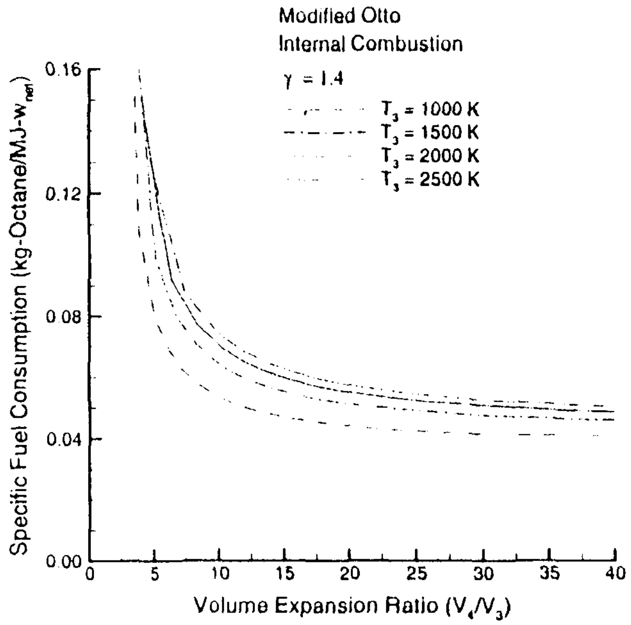

Fig. 11 Specific fuel consumption of modified Otto cycle with internal combustion using liquid air.

The specific air consumption for the modified Otto cycle using internal combustion is shown in Fig. 12. Higher peak temperature reduces the rate liquid air is consumed in this application. It is also apparent that the air consumption rate is not improved very much by using expansion ratios greater than $V_4/V_3 = 20$. The logistical and safety issues of using liquid air for automotive applications need to be carefully considered. One major concern is oxygen enrichment of the cryogen due to boil off of nitrogen during long periods of storage. Means to determine the oxygen-to-nitrogen ratio of the cryogen as it is used may need to be employed to ensure the proper fuel-oxidizer ratio is injected into the cylinder. These issues are important, however, they will not be addressed in any more detail in this study because its primary goal is to determine the energetic potential of hybrid fuel-cryogen propulsion systems. These results are anticipated to be useful for specialty fleet applications where the duty cycle of the vehicles is such that monitoring the oxygen content of the liquid air is not necessary.

EXTERNAL COMBUSTION CYCLE PERFORMANCE – Inert working fluids can be used with the external combustion power cycles. The work output of the processes in the modified Otto and cryogenic Rankine cycles using liquid N_2 are shown in Figs. 13 and 14, respectively. The peak temperature for both of these examples is T_3 = 1000 K. There is no injection work in the externally heated Otto cycle whereas the injection process contributes significantly, at low expansion ratio, to the net work output of the cyrogenic Rankine cycle (ordinate is on left vertical axis). The expansion and exhaust works are identical for both cycles since the expansion process starts and ends at the same peak and exhaust pressures, respectively. The pressure building effect of constant volume heat addition reduces the cryopump work in the modified Otto cycle to about one third that of the Rankine cycle. Over the range of expansion volume shown, the net work of the Rankine cycle exceeds that of the externally heated modified Otto cycle. Due to the increase in cryopump work required for pressurization and the effect of finite clearance volume (V_{clr} = 0.02), however, the net work of the cryogenic Rankine cycle has a maximum at $V_4/V_3 \sim 16$ for this set of operating conditions. Even though the constant volume heat addition process significantly reduces the cryopump work in the modified Otto cycle, the higher specific work output of the Rankine cycle indicates that its expander would be more compact.

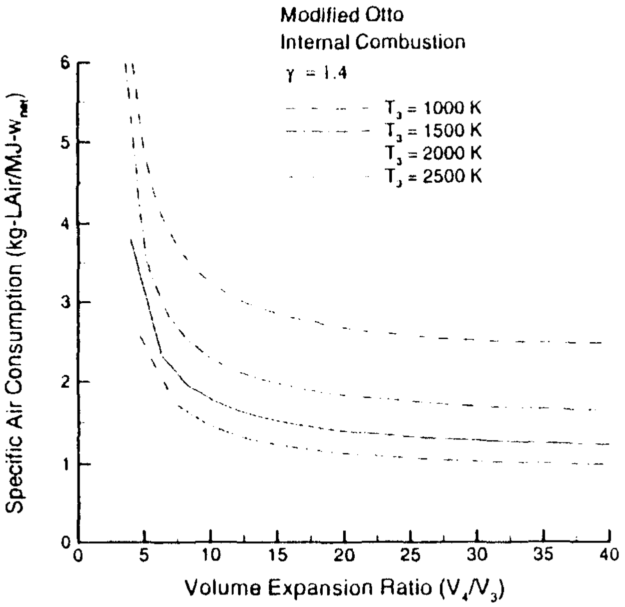

Fig. 12 Specific air consumption of modified Otto cycle with internal combustion using liquid air.

A key advantage of external combustion systems is that many different fuels can be used. Liquid methane (or liquid natural gas) is considered here because it is the cleanest burning of hydrocarbon fuels and the logistical difficulties of its storage may be minimized by the presence of other cryogen handling equipment. The specific fuel consumption of the two externally heated power cycles is shown in Figs. 15 and 16 for liquid methane fuel. The cryogenic Rankine cycle (Fig. 16) is more fuel efficient at expansion ratios less than 16 for any amount of heat addition, however, the external combustion Otto cycle has lower specific fuel consumption at higher expansion. This is due to the large amount of cryopump work required for high pressure (high expansion ratio) operation and the reduction of injection work due to finite clearance volume. Thus the expansion ratio of the expander technology to be employed may dictate which power cycle is best suited for a given application.

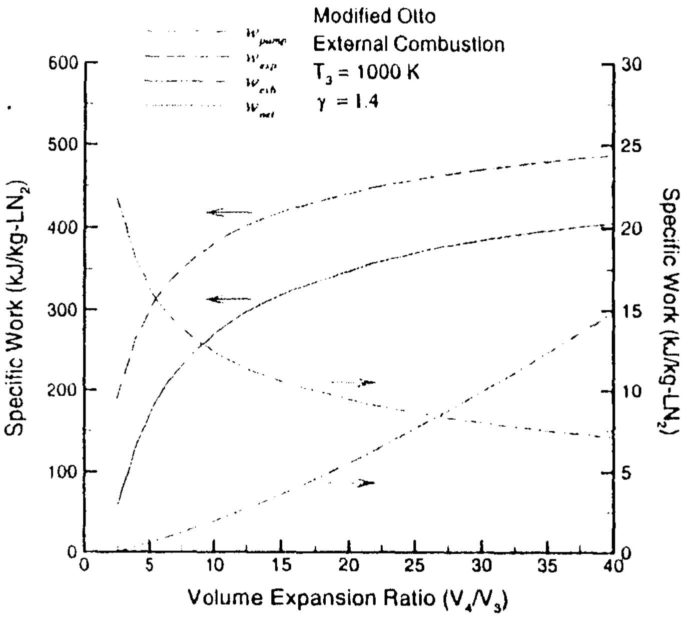

Fig. 13 Specific work of modified Otto cycle with external combustion using liquid N_2.

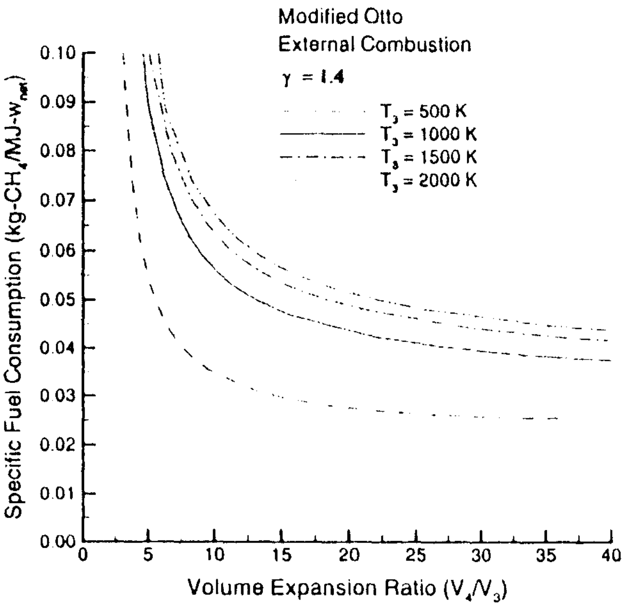

Fig. 15 Specific fuel consumption of modified Otto cycle with external combustion using liquid N_2.

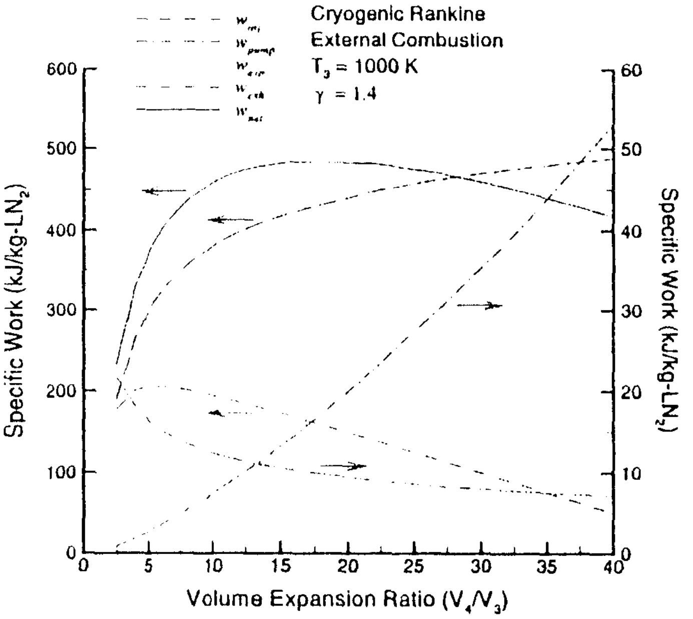

Fig. 14 Specific work of cryogenic Rankine cycle with external combustion using liquid N_2.

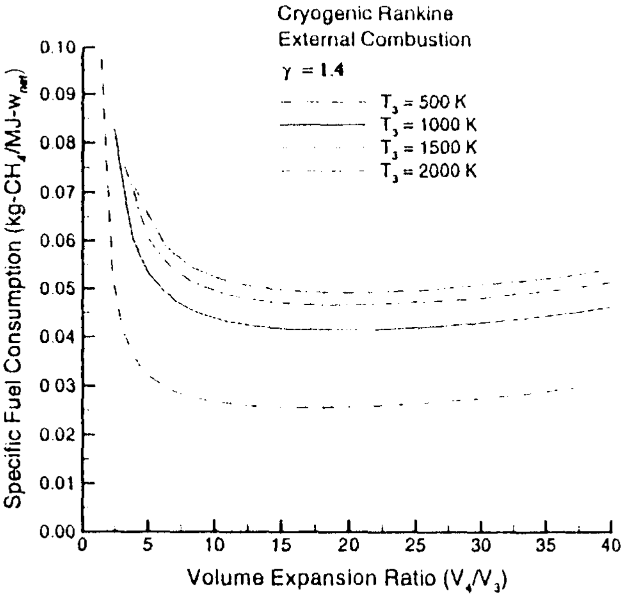

Fig. 16 Specific fuel consumption of cryogenic Rankine cycle with external combustion using liquid N_2.

The specific cryogen consumption (kg-cryogen per unit of net work) of the external heat addition power cycles at peak cycle temperatures of 500 K and 1000 K as a function of expansion ratio are shown in Fig. 17. The liquid N_2 consumption for both cycles is reduced by a factor of two over most of the expansion ratio range shown here by doubling the peak cycle temperature from 500 K to 1000 K. Increasing the peak temperature to 2000 K does reduce the cryogen consumption even further, however, it is not practical to fabricate an automotive firebox that can operate at this high of temperature. The liquid N_2 consumption for the externally heated Otto cycle monotonically decreases with increasing expansion ratio. Whereas the cryogenic Rankine cycle curves have a minimum near $V_4/V_3 \sim 16$ due to increasing cryopump work as previously described.

RANGE AND OPERATING COST – The fuel and cryogen consumption rates for a representative vehicle are determined for realistic operating conditions of the four power cycles under consideration while assuming a mechanical efficiency of 80% for torque transmission to the wheels. The miles per gallon (mpg) estimates are based on a vehicle which has the same road power demand during freeway cruise (7.8 kW) as the 1991 Honda CRX.[15] The standard Otto cycle is assumed to have a compression ratio of 8 and peak cycle temperature of 2000 K. The modified Otto cycle engine with internal combustion has the same peak temperature ($T_3 = 2000$ K) and an expansion ratio of 20. Both of these power plants are assumed to burn octane-air mixtures inside the cylinder as previously described. Mileages for the external combustion systems have been determined using liquid N_2 as the working fluid and either liquid CH_4 or octane as the fuel. The peak temperature in the external combustion systems is limited to 1000 K and their expanders have an expansion ratio of 20. The cryogenic Rankine cycle performance in the absence of combustion heat addition is also determined for the ZEV mode, using the same mechanical efficiencies as before, at a peak cycle temperature of $T_3 = 285$ K and a peak injection pressure of $P_3 = 5.4$ Mpa.

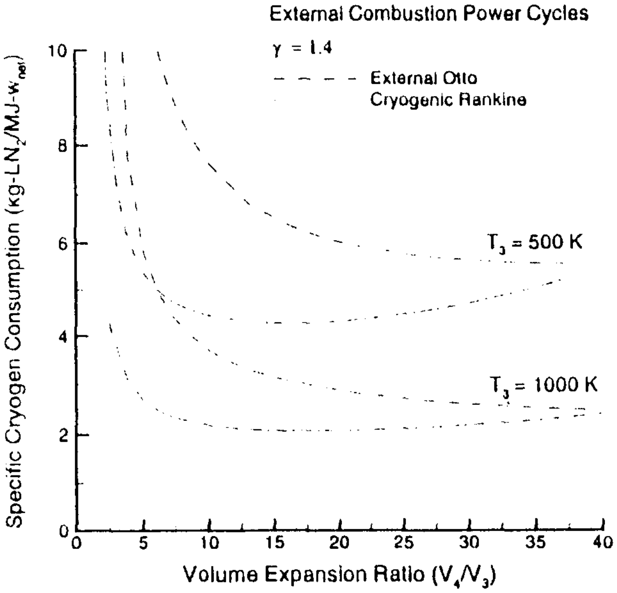

Fig. 17 Specific cryogen consumption of the external combustion cycles using liquid N_2.

The fuel and cryogen mileage of the powerplants, based on the aforementioned assumptions, are compared in Table 2. Operating costs are based on \$1.20 per gallon octane, \$0.60 per gallon liquid CH_4, and \$0.15 per gallon for the liquid air and liquid N_2. Routine maintenance expenses are not included. The model for the standard Otto cycle predicts a lower fuel consumption rate than is typically realized by the reference sedan (Honda CRX) chosen for this study; however, if temperature effects on the heat capacity ratio of the combustion products and more detailed accounting of mechanical and combustion inefficiencies were included, the fuel mileage would be more comparable with what is currently achieved in practice. For the purposes of this study it is sufficient to compare the performance of alternative power cycles to that of the standard Otto to illustrate the potential of unconventional cryogen-fuel hybrid power cycles for automotive transportation.

Table 2. Mileage of hybrid propulsion systems

Cycle	Fuel	Cryogen	Fuel mpg	Cryogen mpg	Net Cost \$ per mi
Standard Otto	C_8H_{18}	N/A	54	N/A	0.022
Internal Otto	C_8H_{18}	Liq Air	83	4.0	0.052
External Otto	CH_4	Liq N_2	63	1.8	0.093
Cryogenic Rankine	CH_4	Liq N_2	62	2.5	0.070
External Otto	C_8H_{18}	Liq N_2	93	1.8	0.096
Cryogenic Rankine	C_8H_{18}	Liq N_2	92	2.5	0.073
Cryogenic Rankine	Ambient Heat	Liq N_2	N/A	0.78	0.192

The improved fuel mileage of the internal combustion engine using cryo-pumped liquid air does not offset the corresponding operating expense, as indicated in the net operating costs per mile of Table 2. The application of combustion heat release to ambient-heated liquid N_2, however, significantly increases its effectiveness as a working fluid. The mileage for the liquid N_2 is double and triple its value for the ZEV mode of operation by externally adding the heating value of hydrocarbon fuel in the modified Otto and Rankine cycles, respectively. The methane-fuel mileage on a per gallon basis for the external heating power cycles is comparable to that of the standard Otto cycle, whereas, it is nearly 70% better when the superheater is fueled with octane. The net operating costs per mile of the external combustion systems using liquid CH_4 or octane do not differ much because the expense of the cryogen is the primary cost factor. The operating costs can be reduced by using a higher peak temperature in the firebox to better utilize the cryogen working fluid. For the same reason the ZEV mode of operation has the highest operating cost of all, however, its liquid N_2 consumption rate can drop by more than a factor of two if multiple reheats and expansions are used[16] or if the expander is capable of quasi-isothermal operation in conjunction with the application of a topping cycle.[9]

DISCUSSION

The dual-fuel cryogen power cycles considered in this study were not optimized to realize their highest potential. The mechanical efficiency of the components were rigorously kept the same to provide a first order comparison of the relative merits of the different hybrid propulsion systems. The combustion efficiency was conservatively estimated and the enhancement of fuel economy due to waste heat recovery was not included. In addition, neither the differences in propulsion system complexity nor the impact of potential purchase price for mass produced vehicles were considered. Furthermore, the challenge of establishing a cryogen distribution infrastructure needs to be weighed against the problems associated with introducing any new technology for automotive transportation. Even though there are many issues yet to be addressed, several general conclusions can be drawn from this study.

Based on the current price of regular grade gasoline in the U.S.A., the results presented in Table 2 indicate that the standard Otto cycle engine has the lowest operating costs. The low peak cycle temperature of the external combustion systems, however, result in lower nitrogen oxide emissions and better longevity of the machinery. Burning methane fuel internally or externally significantly reduces the wear and tear on piston ring seals in the expander and the oil contamination in the crankcase. The potential of making liquid CH_4 (liquid natural gas) readily available for this hybrid vehicle is encouraged by the presence of the cryogen handling equipment. The ability to switch off the fuel flow and operate as a ZEV having better performance than electric vehicles using lead-acid batteries for energy storage is another key advantage. It probably would be more appropriate to compare the performance parameters of these cryogen-fuel hybrid systems with other hybrid vehicle concepts being developed, rather than with the conventional internal combustion engine.

The dual-fuel hybrid systems all have better fuel economy than the standard Otto engine, however, the price of cryogen manufacture and delivery is the primary factor driving up the cost of their operation. The cryogen price of $0.15 per gallon is nearly twice the energy cost of its manufacture (based on $0.05 per kW-h) using currently available liquefaction technology. More favorable off-peak electrical power rates and advances in air manufacture processes would reduce the cost of liquid air products. If there is a need to operate in urban areas having significant restrictions on tail pipe pollutants, then the ultra-low emission capability of the dual-fuel hybrid propulsion systems may outweigh the indicated increase in fuel-cryogen operating expense. In addition, the reduction in maintenance needs for cleaner burning combustion systems may completely offset the higher fuel operating costs over the useful lifetime of the vehicles.

SUMMARY

A theoretical performance model for reciprocating engines has been applied to the standard Otto cycle and three other power cycles involving the use of fuel and cryogens. Performance trends have been illuminated which indicate the effect of many operating parameters on the net work output of the cycles. Fuel and cyrogen consumption rates have been determined and the potential mileage and operating costs of a hybrid vehicle have been estimated. Operating costs of ~$0.05 per mile have been estimated for vehicles having fuel (gasoline) and cryogen (liquid air) consumption rates of 80-90 mpg and ~4 mpg, respectively. This fuel mileage is ~50% higher than the theoretical performance of the standard Otto cycle.

The hybrid propulsion systems discussed here are based on readily available technologies which can probably be integrated into a mass produced automobile in a relatively short time frame. Their performance potential is comparable to modern automobiles and the refueling process would be similar to current practice. The biggest obstacle to the ultimate implementation of these cryogen propulsion systems is the development of a cryogen distribution infrastructure similar to that of current gasoline stations. The low technological risk in developing these cryogenic propulsion systems and potential environmental benefits of their widespread use encourage further studies along this line of research.

ACKNOWLEDGMENTS

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